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HEAT-TRANSMITTING TUBES, (U)

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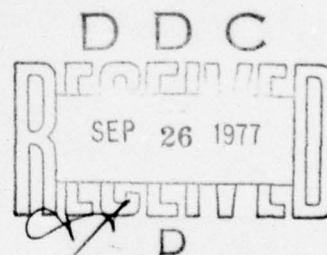
FOREIGN TECHNOLOGY DIVISION



HEAT-TRANSMITTING TUBES

by

L. L. Vasil'yev, S. V. Konev



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HEAT-TRANSMITTING TUBES

By: L. L. Vasil'yev, S. V. Konev

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U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	<i>А а</i>	A, a	Р р	<i>Р р</i>	R, r
Б б	<i>Б б</i>	B, b	С с	<i>С с</i>	S, s
В в	<i>В в</i>	V, v	Т т	<i>Т т</i>	T, t
Г г	<i>Г г</i>	G, g	У у	<i>У у</i>	U, u
Д д	<i>Д д</i>	D, d	Ф ф	<i>Ф ф</i>	F, f
Е е	<i>Е е</i>	Ye, ye; E, e*	Х х	<i>Х х</i>	Kh, kh
Ж ж	<i>Ж ж</i>	Zh, zh	Ц ц	<i>Ц ц</i>	Ts, ts
З з	<i>З з</i>	Z, z	Ч ч	<i>Ч ч</i>	Ch, ch
И и	<i>И и</i>	I, i	Ш ш	<i>Ш ш</i>	Sh, sh
Й й	<i>Й й</i>	Y, y	Щ щ	<i>Щ щ</i>	Shch, shch
К к	<i>К к</i>	K, k	Ъ ъ	<i>Ъ ъ</i>	"
Л л	<i>Л л</i>	L, l	Ы ы	<i>Ы ы</i>	Y, y
М м	<i>М м</i>	M, m	Ь ь	<i>Ь ь</i>	'
Н н	<i>Н н</i>	N, n	Э э	<i>Э э</i>	E, e
О о	<i>О о</i>	O, o	Ю ю	<i>Ю ю</i>	Yu, yu
П п	<i>П п</i>	P, p	Я я	<i>Я я</i>	Ya, ya

*ye initially, after vowels, and after ъ, ъ; e elsewhere.
 When written as ё in Russian, transliterate as yë or ë.
 The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

GREEK ALPHABET

Alpha	A	α	α	Nu	N	ν
Beta	B	β		Xi	Ξ	ξ
Gamma	Γ	γ		Omicron	Ο	ο
Delta	Δ	δ		Pi	Π	π
Epsilon	E	ε	ε	Rho	Ρ	ρ ϑ
Zeta	Z	ζ		Sigma	Σ	σ ς
Eta	H	η		Tau	Τ	τ
Theta	Θ	θ	θ	Upsilon	Υ	υ
Iota	I	ι		Phi	Φ	φ ϕ
Kappa	K	κ	κ	Chi	Χ	χ
Lambda	Λ	λ		Psi	Ψ	ψ
Mu	M	μ		Omega	Ω	ω

RUSSIAN AND ENGLISH TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	\sin^{-1}
arc cos	\cos^{-1}
arc tg	\tan^{-1}
arc ctg	\cot^{-1}
arc sec	\sec^{-1}
arc cosec	\csc^{-1}
arc sh	\sinh^{-1}
arc ch	\cosh^{-1}
arc th	\tanh^{-1}
arc cth	\coth^{-1}
arc sch	sech^{-1}
arc csch	csch^{-1}
<hr/>	
rot	curl
lg	log

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PAGE 1

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Heat-transmitting tubes.

L. L. Vasil'yev, S. V. Konev

Pages 1-43.

HEAT-TRANSMITTING TUBES.

FTD-ID(RS)T-0165-77

L. L. Vasiliev, S. V. Konev

edited by academician ^{of} the A.S. of the B.S.S.R., A. V. Lykova

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Minsk 1972.

Page 2.

Vasiliev L. L., Konev S. V. ~~heat~~ heat-transmitting tubes. Minsk, "science and engineering", 1972, page 152.

In the book are described the different forms of low-temperature thermal ^{tubes} ~~ducts~~. Are presented the theoretical principles ~~in them~~ of the processes taking place ^{in them} ~~heat- and mass exchange~~. Considerable attention is devoted to the investigation of the transport properties of capillary-porous cores, to the ~~application~~ use of thermal ducts for cooling radio-electronics equipment. Is given the characteristic of the different constructions of the controlled thermal ^{tubes} ~~ducts~~, such, as ducts with the presence of ~~the~~ residual ~~of~~ non-condensable gas, centrifugal thermal ^{tubes} ~~ducts~~ etc.

Tables 3, figures 37, bibliography - 114 names.

It is designed for the workers of scientific research institutes, design organizations, design bureau^s, the technical personnel of the industry, graduate students and students of schools of higher education.

Page 3.

Designations.

P - pressure, N/m^2 ;

Q - power, W ;

q - heat transfer rate, W/m^2 ;

C - the constant of proportionality;

r' - ^{latent} heat of vaporization, J/kg ;

σ - surface tension, N/m ;

μ - ^{viscosity} ~~dynamic viscosity~~, $kg/m \cdot s$;

ρ - density, kg/m^3 ;

ν - μ/ρ ;

δ - thickness, m;

h_m - the height ~~altitude~~ of capillary absorption, m;

ϕ - slope angle;

K - permeability, m^2 ;

K_1 - $1/K$, $1/m^2$;

A ~~i~~ cross-sectional area, m^2 ;

L , l - length, m;

r - radius, m;

j - the mass flow, kg/s;

θ - the angle of wetting;

τ - time, s;

ψ - capillary potential, m^2/s^2 ;

T, t - temperature;

h - enthalpy, J/kg ;

b - the width of core, m ;

α - the coefficient of heat exchange, $\text{W/m}^2 \cdot \text{deg}$;

α' - the thermal-expansion coefficient, deg^{-1} ;

Π - porosity;

λ_{eff} - ~~effective~~ effective thermal conductivity of core;

g_0 - the free-fall acceleration, which corresponds to the conditions of the testing of specimen ~~sample~~, m/s^2 ;

g - the free-fall acceleration, which corresponds to test conditions of thermal ~~test~~ ^{tube}, cm/s^2 ;

v - linear speed, m/s ;

St - Stanton's criterion;

Pr - Prandtl number;

$N_p = \frac{\rho \mu g}{p \tau}$ - the parameter of pressure;

Re - Reynolds number;

Ra - Rayleigh's criterion.

~~Figure 4.~~

Indices:

ж - liquid;

к - condenser ~~capacitor~~;

и - evaporator ~~vaporiser~~;

τ - tube;

a - the adiabatic zone of ~~duct~~ ^{tube};

об - ~~specimen~~ sample;

cp - average;

max. - maximum;

min - minimum;

опт - optimum;

ст - wall;

нас - saturation;

пуз - bubble;

п - ~~liquid~~ vapor;

кап - capillary;

общ - common, ~~general, total, i.e., common/general/total~~;

B.O - pressurization volume.

~~Page 5.~~

The introduction

Let us examine the works, dedicated to the study of the thermal ~~ducts~~^{tubes} and steam chambers in which was utilized as heat-transfer agent the liquid, possessing ~~the~~^a low coefficient of thermal conductivity and ~~the~~ low boiling point, in essence ~~these are~~^{these are} thermal ~~ducts~~^{Tubes}, utilized in the range of temperatures below 500°K, filled by such heat-transfer agents as water, alcohol, ammonia, acetone, N₂O₄, Freon, liquid oxygen, nitrogen, hydrogen etc. Let us ~~also~~^{call} them ~~the~~ low-temperature thermal ~~ducts~~^{tubes} and ~~the~~ steam chambers.

Thermal ~~ducts~~^{tubes} won acceptance at present in a ~~series~~^{number} of ~~the~~ branches of industry. High-temperature ~~ducts~~^{tubes} it is assumed to ~~be~~ use extensively in power engineering for the ~~cooling~~ derivation of thermal energy from nuclear and isotopic reactors, the creation of thermionic-emitting and thermoelectric generators, in metallurgical and electronic industry. It is published at present more than a thousand^{on} of articles and patents ~~along~~ high-temperature thermal ~~ducts~~^{tubes}.

Low-temperature thermal ~~units~~^{tubes} obtained development from 1967 and they are utilized in electronic industry for cooling oscillator tubes, travelling-wave tubes, klystrons etc.; in power engineering - for blade cooling of turbines, generators, engines; in machine-tool industry - for cooling cutters, ~~cylinders~~^{milling tools} etc.; in light industry - for the production of pressure cookers, rods for the frying of shashlik, hens, preparation of biscuits etc.; in ~~cooling~~^{refrigeration} industry - for cooling the ~~volumes~~^{storage spaces} of everyday coolers; in the medical and biological industry - for thermostatic control and cooling ~~the~~^{of} individual sections of ~~the~~^{tissue} human ~~labor~~, blood, sperm, etc.

Page 6.

The thermal ~~units~~^{tubes} and ~~the~~ steam chambers have a series of advantages in comparison with the traditional system elements of ~~the~~ heat transfer, for example, ~~by the~~ circulation heat exchangers: they do not have movable parts, are noiseless, do not require energy consumption for the pumping of heat-transfer agent from zone of condensation into the zone of evaporation, they possess low thermal resistance in comparison with the ~~metal rods~~^{metal rods} of the same geometric parameters and have low weight.

Is known at present many different types of ~~the~~ thermal ~~units~~^{tubes} and steam chambers [1-36]. Their classification it is possible to

realize ^{by} ~~in~~ a series of ~~simple~~ criteria, such as the temperature range of use, the degree of a change in their thermal resistance, the method of the transfer of heat-transfer agent from the zone of condensation into the zone of evaporation, the dimensional characteristics of housing and elements etc.

The classification of the types of thermal ~~ducts~~ ^{tubes according to} ~~over~~ their working temperature range ~~is~~ is following:

- 1) high-temperature thermal ~~ducts~~ ^{tubes} ($1200^{\circ}\text{K} \leq T \leq 3000^{\circ}\text{K}$);
- 2) the thermal ~~ducts~~ ^{tubes} of ~~the~~ moderate temperature range ($300^{\circ}\text{K} \leq T \leq 1200^{\circ}\text{K}$);
- 3) the low-temperature or cryogenic thermal ~~ducts~~ ^{tubes} ($1^{\circ}\text{K} \leq T \leq 300^{\circ}\text{K}$).

Depending on geometric dimensions it is possible to distinguish thermal ~~ducts~~ ^{tubes} whose length L substantially exceeds their diameter ($L/D \gg 10$) (Fig. ^{1a)} ~~1a~~, and steam chambers whose $L/D \leq 10$ (Fig. 1b). It is natural that the form of the thermal ~~ducts~~ ^{tubes} and steam chambers can be different.

Large interest ~~they~~ represent the thermal ~~ducts~~ ^{tubes} of

~~alternating~~ variable thermal resistance or the controlled thermal ^{tubes} ~~tubes~~ (thermal ^{tubes} ~~tubes~~ with the use of ^a ~~the~~ residual ~~of~~ non-condensable gas, electrical, ultrasonic, magnetic fields and centrifugal fields, thermal diodes and triodes etc.) [25, 31, 34, 35, 38, 84, 85].

The thermal ^{tubes} ~~tubes~~, utilized in electronic industry for cooling high-voltage oscillator tubes, combine in themselves properties, it would seem, ^{that are} incompatible: they have negligible thermal resistance and good dielectric properties, ~~that~~ it is possible to name ^{them} high-voltage thermal ^{tubes} ~~tubes~~ [105].

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Which parameters of thermal ^{tubes} ~~tubes~~ are most important? First, the maximum value of the heat output, transferred along ^{tube} ~~tube~~. It is determined either by the onset of the crisis of boiling or by the gas-dynamic closing of steam channel, or by the limited productivity of capillary pump. In the second place, the thermal resistance of ^{tube} ~~tube~~ (it depends on the thermophysical properties of the core and heat-transfer agent, presence of the residual ~~of~~ non-condensable gas). Thirdly, the coefficient of heat exchange on the external surface of evaporator ~~evaporator~~ and condenser ~~condenser~~.

One of the most promising ways of a reduction in the thermal resistance is the realization of the induced convection of liquid in the zone of evaporation and condensation with the aid of, for example, different fields (magnetic, electrical, ultrasonic, temperature, gravitational, centrifugal etc.) and a decrease in the thickness of the fluid film and porous core. In this case for the supply of the necessary amount of liquid into the zone of evaporation from the zone of condensation, are utilized the supplementary arteries, which can be ~~arrange~~ located along the ~~axis~~ ^{tube} axis of thermal ~~duct~~ (Fig. 2) [31, 32, 43, 105].

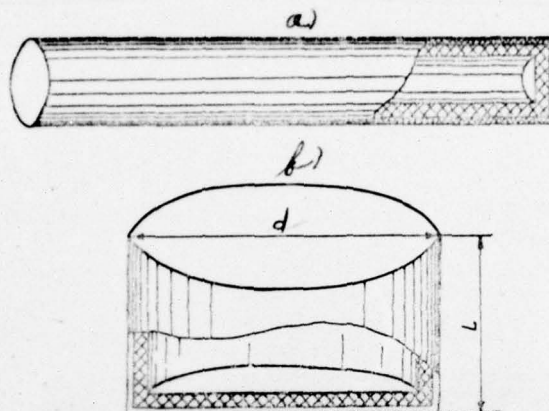


Fig. 1. Thermal tube (a) and ~~the~~ steam chamber (b).

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The selection of heat-transfer agent for the thermal ^{tubes} ~~ducts~~ and ~~the~~ steam chambers is realized on the basis of the following requirements: 1) the maximum value of the coefficient of surface tension σ and ~~the~~ good wetting properties for providing the necessary capillary pressure head ΔP_{cap} ~~of 2-3 mm~~ 2) low ductility ~~roughness, viscosity~~ in order to facilitate the conditions of the transport of liquid along porous core; 3) the high ^{latent} heat of vaporization for achievement of the maximum heat removal from ^a ~~the~~ unit of surface; 4) high thermal conductivity in order to decrease the thermal resistance of the core, filled ^{with} ~~by~~ liquid; 5) the operating range of temperatures ~~at~~ must be located ⁱⁿ ~~at~~ temperature range between the triple and critical point; 6) the high density ~~of~~ ^{the} ~~in~~ in order to decrease the process of the interaction of flow ^{of vapor} ~~with~~ ~~the~~ with liquid; 7) inertness with respect to the material of core and housing of thermal duct.

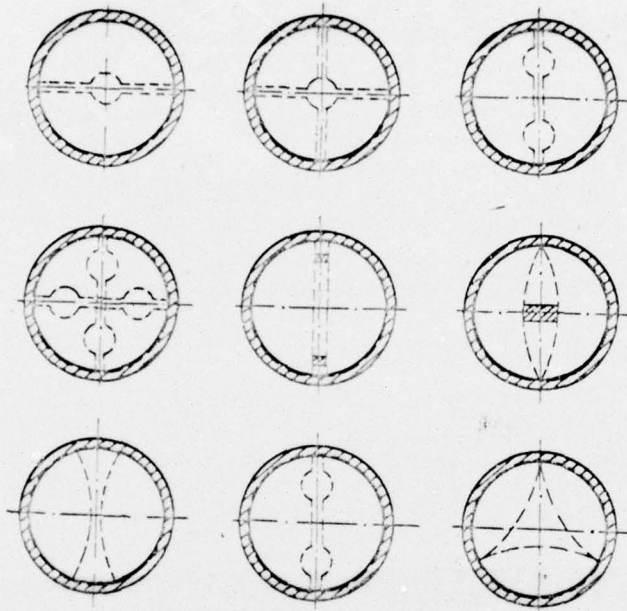


Fig. 2. Forms of arterial thermal ^{tubes} ~~elements~~ [36].

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Chapter 1

THEORETICAL PRINCIPLES OF THE WORK OF THERMAL TUBES.

1. High-temperature or "isothermal" thermal ~~tubes.~~ ^{tubes.}

Known at present theoretical works ~~along~~ ^{on} thermal ~~tubes~~ ^{tubes} are based on the assumption that the thermophysical and thermodynamic properties of heat-transfer agent and capillary-porous core are constant and do not depend on temperature. In essence they describe high-temperature thermal ~~tubes~~ ^{tubes.}

Cooling ^{of} the condenser ~~apparatus~~ ^{of} high-temperature thermal ^{tubes} ~~tubes~~ is realized most frequently by emission ~~radiation~~ into the environment and by means of the thermal conductivity of the rarefied gas. For them is characteristic the small area of heating (evaporator) ~~apparatus~~ and the substantially large area of cooling

(condenser ~~capacitor~~). As heat-transfer agent usually is utilized any metal in the molten state.

The porous core of high-temperature thermal ~~ducts~~ ^{tubes} usually has ~~low~~ ^{low} hydraulic friction and high thermal conductivity. The fundamental performance characteristic of thermal ~~duct~~ ^{tube} is the maximum value of the transferred heat output in gravitational field and under conditions of weightlessness and the absolute temperature, ~~at~~ ^{at} which ~~works~~ ^{the tube works}.

The maximum value of the heat output, transferred along thermal duct under stationary conditions, is equal to the sum of the heat output, transferred by convection and ~~the~~ thermal conductivity:

$$Q_{\max} = Q_{\text{conv}} + Q_{\text{cond}} \quad (1.1)$$

The heat transfer by thermal conductivity in axial direction in the housing of ~~duct~~ ^{tube} and porous core it is possible to disregard in comparison with transmission ^{by} convection; therefore

$$Q_{\max} = h \rho_n U_x = j_{\text{ж. max}} r', \quad (1.2)$$

where h is enthalpy; ρ_n - density ^{of vapor} ~~of liquid~~; U_x - the speed ^{of vapor} ~~of liquid~~ in axial direction; $j_{ж, max}$ ~~the~~ the maximum fluid flow; r' - heat of vaporization.

~~Page 10.~~

The fluid flow ~~of~~ $j_{ж, max}$ - ^{over} ~~the~~ capillary-porous core is determined on the basis of the transport properties of core ^{based on} ~~the~~ the transfer of the selected liquid with the aid of capillary forces under the action of the gradient of capillary potential.

For the stationary working conditions of tube, it is necessary, ^{so that} ~~under~~ pressure change in the closed loop within the tube ~~is~~
 $\Sigma P = 0$, i.e.,

$$(P_{n(n)} - P_{n(n)}) + (P_{n(n)} - P_{ж(n)}) + (P_{ж(n)} - P_{ж(n)}) + (P_{ж(n)} - P_{n(n)}) = 0. \quad (1.3)$$

At the same time it is necessary, ^{that} in order ~~to~~

$$\Delta P_n + \Delta P_{ж} \leq \Delta P_{кап} \quad (1.4)$$

or

$$\Delta P_{ж(тр.)} + \Delta P_{ж(вс)} + \Delta P_{n(аэр.сопр.+инерция)} \leq \Delta P_{кап}. \quad (1.5)$$

^{air resistance}
^{inertia}

According to the equation ^{σ} Laplace - Young, on the curved surface of ~~section~~ ^{interface} liquid - vapor the pressure differential is equal to

$$\Delta P = \sigma \left(\frac{1}{R'} + \frac{1}{R''} \right), \quad (1.6)$$

where R' and R'' are the radii of curvature, which describe the three-dimensional surface of the meniscus of liquid.

If liquid wets ^{the} core, the angle of contact ^{is} less than 90° . Let us assume that the interface ^{liquid - vapor} in condenser ~~evaporator~~ and evaporator ~~evaporator~~ is spherical ($R' = R''$) ^{and} is described by radii of R_R and R_H , ~~then~~ then

page 2°

$$P_{n(n)} - P_{n(n)} = \frac{2\sigma}{R_n}, \quad (1.7)$$

$$P_{n(n)} - P_{n(n)} = \frac{2\sigma}{R_n}, \quad (1.8)$$

$$\Delta P_{\text{cap}} = \frac{2\sigma}{R_n} - \frac{2\sigma}{R_n}. \quad (1.9)$$

Under the optimum conditions of the work of ~~the~~ ^{tubes for} radius of interface vapor - liquid in condenser ~~approaches~~ approaches infinity $R \rightarrow \infty$, i.e., there is a thin film of liquid on porous surface.

~~Page 44.~~

Then

$$\Delta P_{\text{cap}} = -\frac{2\sigma}{R_n} = -\frac{2\sigma}{R_{\min}}, \quad (1.10)$$

$$R_{\min} = \frac{2\sigma}{\rho g h_{\max}}, \quad (1.11)$$

where ~~the~~ h_{\max} ~~are~~ ^{height} the maximum ~~altitude~~ of capillary elevation.

The capillary pressure in gravitational field and the potential of ~~the~~ transfer are defined as

$$\psi = \frac{2\sigma}{\rho_m} \cdot \frac{\cos \theta}{r_{min}} + gh \sin \alpha, \quad (1.12)$$

$$\Delta P_{kan} = \frac{2\sigma \cos \theta}{r_{min}} \pm \rho gh \sin \alpha. \quad (1.13)$$

The pressure differential in liquid phase. Force³ of inertia in the liquid phase of thermal ~~ducts~~^{tubes} usually they disregard and is examined ^a ~~the~~ laminar viscous fluid flow in pores, which obeys the law of ~~the~~ Darcy:

$$\Delta P_m = - \frac{\mu_m j_m L_{eff}}{\rho_m K S}. \quad (1.14)$$

In this expression they use the effective length of thermal ~~duct~~^{tube}

$$L_{\text{эф}} = l_{\text{a.3}} + \left(\frac{l_{\text{н}} + l_{\text{к}}}{2} \right). \quad (1.15)$$

This is caused by the fact that in the zone of evaporation and condensation is assumed ~~as~~ the presence of the uniform radial flow of mass as a result of evaporation and condensation [107].

The pressure differential due to the forces of gravitation is equal to

$$\Delta P_{\text{г.р}} = -g\rho_{\text{ж}}L \sin \theta. \quad (1.16)$$

The account of the forces of friction and gravitation gives the total pressure differential in liquid phase

$$\Delta P_{\text{ж}} = -\frac{\mu_{\text{ж}} j_{\text{ж}} L_{\text{эф}}}{\rho_{\text{ж}} K S} - g\rho_{\text{ж}}L \sin \theta. \quad (1.17)$$

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If we disregard the pressure differential in vapor phase, then the maximum fluid flow ^{over} ~~on~~ capillary-porous core it is possible to find from formula

$$j_{\text{ж}} = \left(\frac{KS}{L_{\text{эф}}} \right) \left(\frac{\sigma \cos \alpha \rho_{\text{ж}}}{\mu_{\text{ж}}} \right) \left(\frac{2}{R_{\text{min}}} - \frac{g \rho_{\text{ж}} L \sin \theta}{\sigma \cos \alpha} \right), \quad (1.18)$$

where the angle θ ~~is~~ characterizes the ~~angle of~~ inclination of thermal

~~to~~^{tube} line of horizon. Usually it lies ~~in~~ⁱⁿ limits of 0-180°. α is the angle, formed by the surface of liquid and by solid surface in capillaries during their wetting; for the wetting liquids of $\alpha \leq 90^\circ$.

If one assumes that ~~all~~^{all the} heat, applied to thermal ~~data~~^{tube} is expended ~~on~~^{ed} on the process of phase transition, it is possible to determine the heat output, transferred by the ~~data~~^{tube}:

$$Q = j_{\text{m}} r' = r' \left(\frac{KS}{L_{\text{a}\phi}} \right) \left(\frac{\sigma \cos \alpha \rho_{\text{m}}}{\mu_{\text{m}}} \right) \times \\ \times \left(\frac{2}{R_{\text{min}}} - \frac{g \rho_{\text{m}} L \sin \theta}{\sigma \sin \alpha} \right). \quad (1.19)$$

For $g = 0$

$$Q = r' \left(\frac{2\sigma \rho_{\text{m}}}{\mu_{\text{m}}} \right) \left(\frac{KS}{R_{\text{min}} L_{\text{a}\phi}} \right). \quad (1.20)$$

If we instead of R_{min} utilize h_{max} , then

$$Q = \frac{KSr'g\rho_{\kappa}^2}{\mu_m L_{\Phi}} (h_{max} - L \sin \theta). \quad (1.21)$$

For $q = 0$

$$Q = \frac{KS\rho_{\kappa}^2 r' h_{max} g}{\mu_m L_{\Phi}}. \quad (1.22)$$

The process of the motion of liquid in porous body under the action of the gradient of total pressure more correctly is described by the generalized law of ~~arcy~~ ^darcy. In ~~its~~ one-dimensional case it is possible to express ₁ as follows:

$$i_m = -K(\theta) \text{grad } \Phi, \quad \Phi = \psi \pm h. \quad (1.23)$$

However, in real thermal tubes by no means always it is possible to accept the condition that the flow in core is one-dimensional.

~~Page 12.~~

Specifically, for the porous cores of ~~the~~ small thickness when flow ^{of vapor} moves at a high speed, it is necessary to consider its interaction with liquid near surface (wave formation on surface) which shows up in speed distribution of liquid ^{over} ~~according to~~ the section of core. Therefore it is necessary to distinguish at least two velocity component^s of liquid

$$U_x = -K \frac{\partial P}{\partial x}, \quad U_y = -K \frac{\partial P}{\partial y}. \quad (1.24)$$

The description of the process of the motion of liquid in the core when its thickness $a \ll R$, can be manufactured with the aid of the equation of Poisson

$$\frac{\partial U_x}{\partial x} + \frac{\partial U_y}{\partial y} = -K \left(\frac{\partial^2 P}{\partial x^2} + \frac{\partial^2 P}{\partial y^2} \right). \quad (1.25)$$

However, ^{during} ~~the~~ the calculation of cores ~~by~~ this effect usually ^{they} disregard. The ~~common~~ general ~~equation~~ equation, which describes the mass transfer in porous body, is [108]

$$\frac{\partial \theta}{\partial \tau} + \tau' \frac{\partial^2 \theta}{\partial \tau^2} = \text{div} (a_m \text{grad } \theta), \quad (1.26)$$

where the $a_m = K \frac{\partial \Phi}{\partial \theta}$ - the coefficient of hydraulic diffusion;

$$\Phi = h + \int_{P_0}^P \frac{dP}{\rho_m}. \quad (1.27)$$

The use of equation (1.26) is especially important for the examination of the processes of thermal shock in thermal ~~tubes~~ ^{tubes}, which are located in gravitational field, when applied heat flux produces the intense evaporation of liquid and ^{the} condensate ~~is~~ moves over unsaturated porous core at the final speed.

In a number of cases when the term $\tau' \frac{\partial^2 \theta}{\partial \tau^2}$ can be disregarded, equation (1.26) can be simplified and is presented in the form

$$\frac{\partial \theta}{\partial \tau} = \text{div} [a_m \text{grad } \theta] + \frac{\partial K(\theta)}{\partial x}. \quad (1.28)$$

~~Page 14.~~

For the solution to this equation in the case of the unsteady process of the motion of liquid along unsaturated porous core, it is necessary to know the values of $a_m(0)$ and $K(0)$ for ~~a~~ *the* ~~specific~~ ~~material~~ porous material. Under the stationary working conditions of thermal ~~substance~~ *tube* a_m and K are constants.

Nonlinear differential equations (1.26), (1.28) present significant difficulties for solution not only by analytical, but also ^{by} numerical ~~methods~~ *both* as a result of the powerful nonlinear dependence of $a_m(0)$ and $K(0)$, and as a result of the large difference in the speed of absorption ~~at~~ ^{at t_{10}} initial and that which follow points in time. In work [109] ^{to} ₁ given numerical solution of the

two-dimensional problem of mass transfer in porous media.

Frequently as an approximation ^{is} is utilized the exponential dependence of $a_m(\theta)$. In this case equation (1.28) can be rewritten as

$$\frac{\partial \theta}{\partial \tau} = \frac{\partial}{\partial x} \left[C e^{\theta} \frac{\partial \theta}{\partial x} \right] \quad (1.29)$$

and, therefore, to present as

$$\frac{\Delta x^2}{\Delta \tau} (\theta_j^{r+1} - \theta_j^r) = \exp(\theta_{j+1}^r) (\theta_{j+1}^r - \theta_j^r) - \exp(\theta_{j-1}^r) (\theta_j^r - \theta_{j-1}^r), \quad (1.30)$$

which makes it possible to calculate θ_j^{r+1} .

For the description of the process of ~~the~~ mass transfer in the porous core of thermal ~~and~~ ^{tube it is} sufficient to use the equations of the

filtration transfer of ^a liquid in which ~~the~~ $a_m(\theta)$ and $K(\theta)$ are accepted as constants [110]:

$$\operatorname{div} \left[\frac{\rho K}{\eta} (\nabla P + \rho \nabla h) \right] = \Pi \frac{\partial \rho}{\partial \tau}. \quad (1.31)$$

This equation for the case of homogeneous liquid in isotropic porous material in cylindrical coordinates takes the form

$$\begin{aligned} & \frac{1}{r} \cdot \frac{\partial}{\partial r} \left[\frac{r \rho K}{\eta} \left(\frac{\partial P}{\partial r} + \rho \frac{\partial h}{\partial r} \right) \right] + \\ & + \frac{1}{r^2} \cdot \frac{\partial}{\partial \psi} \left[\frac{\rho K}{\eta} \left(\frac{\partial P}{\partial \psi} + \rho \frac{\partial h}{\partial \psi} \right) \right] + \\ & + \frac{\partial}{\partial x} \left[\frac{\rho K}{\eta} \left(\frac{\partial P}{\partial x} + \rho \frac{\partial h}{\partial x} \right) \right] = \Pi \frac{\partial \rho}{\partial \tau}. \end{aligned} \quad (1.32)$$

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The pressure differential in vapor phase. As ~~in~~ ⁱⁿ ~~in~~ ⁱⁿ drop in the pressure ~~of~~ ΔP_n in the vapor phase of thermal ~~dist~~ ^{tube} occurs as a result of the presence of the forces of friction and inertia during

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of vapor. Figure 3, a, b, c gives ~~the curves of~~ ^{the curves of} the pressure ^{drop} in the vapor phase of high-temperature ~~tubes~~ ^{tubes} depending on the velocity of vapor ~~U_x~~ and geometric dimensions of ~~tube~~ ^{tube}.

Flow ~~in~~ ^{of vapor} in evaporator ~~and condenser~~ ^{and condenser} of ~~tube~~ ^{tube} can be characterized by the radial velocity ~~of~~ ^{of} U_r and axial U_x . In the heat-insulated ^{te} ~~part~~ ^{part} of the ~~tube~~ ^{tube} we examine ~~only~~ ^{only} axial velocity ~~of~~ ^{of} U_x .

Let us examine thermal ~~in~~ ^{tube} ~~in~~ ⁱⁿ the form of cylinder. Navier-Stokes equation ~~in~~ ⁱⁿ cylindrical coordinates in steady-state operating conditions of tube takes the form

$$\begin{aligned}
 U_r \frac{\partial U_x}{\partial r} + U_x \frac{\partial U_x}{\partial x} &= -\frac{1}{\rho} \cdot \frac{\partial P}{\partial x} + \\
 &+ \nu \left[\frac{1}{r} \cdot \frac{\partial}{\partial r} \left(r \frac{\partial U_r}{\partial r} \right) + \frac{\partial^2 U_x}{\partial x^2} \right], \\
 U_r \frac{\partial U_r}{\partial r} + U_x \frac{\partial U_r}{\partial x} &= -\frac{1}{\rho} \cdot \frac{\partial P}{\partial r} + \\
 &+ \nu \left[\frac{\partial}{\partial r} \left(\frac{1}{r} \cdot \frac{\partial}{\partial r} (r U_r) \right) + \frac{\partial^2 U_r}{\partial x^2} \right]. \quad (1.33)
 \end{aligned}$$

Equation of continuity

$$\frac{\partial (r U_x)}{\partial x} + \frac{\partial (r U_r)}{\partial r} = 0. \quad (1.34)$$

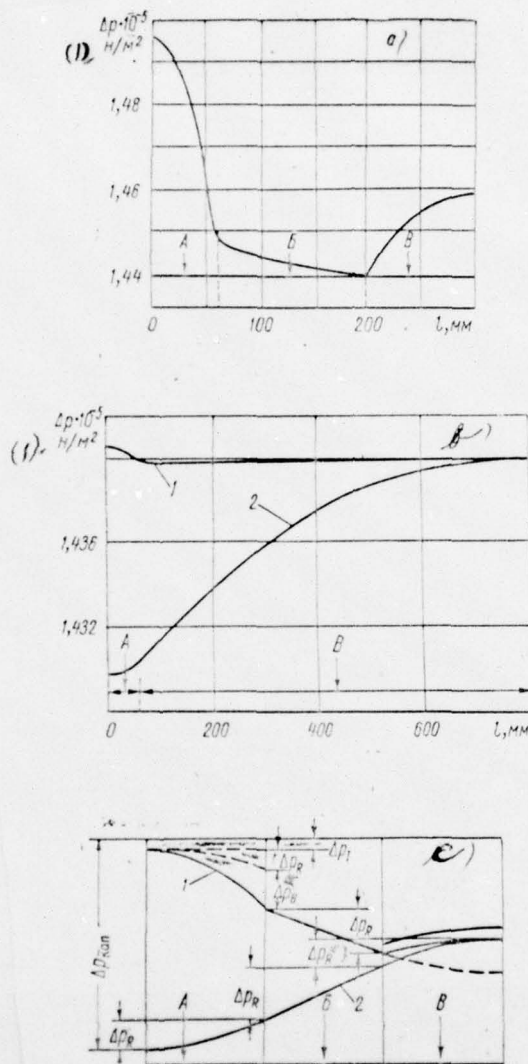


Fig. 3. Dependences of a pressure drop ^{of vapor} ~~for~~ for tube $L = 300 \text{ mm}$; $L = 800 \text{ mm}$ (a, b) and ^{theoretical} ~~calculated~~ curves (c) [36].

Key: (1) N/m^2 .

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If one takes into account, that the length of thermal ~~tube~~ ^{tube} is considerably greater than ~~a~~ radius, tube has invariable geometry and ~~the~~ constant thermophysical properties of the material of heat-transfer agent, then the given system of equations can be significantly simplified:

$$\frac{\partial P}{\partial x} = -\rho \left(U_r \frac{\partial U_x}{\partial r} + U_x \frac{\partial U_x}{\partial x} \right) + \dots$$

$$+ \mu \left(\frac{1}{r} \cdot \frac{\partial U_x}{\partial r} + \frac{\partial^2 U_x}{\partial r^2} + \frac{\partial^2 U_x}{\partial x^2} \right), \quad (1.35)$$

$$\frac{\partial P}{\partial r} = -\rho \left(U_r \frac{\partial U_r}{\partial r} + U_x \frac{\partial U_r}{\partial x} \right) +$$

$$+ \mu \left(\frac{\partial^2 U_r}{\partial r^2} + \frac{\partial^2 U_r}{\partial x^2} + \frac{1}{r} \cdot \frac{\partial U_r}{\partial r} - \frac{U_r}{r^2} \right), \quad (1.36)$$

$$\frac{\partial (r U_x)}{\partial x} + \frac{\partial (r U_r)}{\partial r} = 0. \quad (1.37)$$

Boundary conditions for the solution of this system of equations can be written as follows:

$$U_{r(0)} = -U_R,$$

$$r = R, U_x = 0, U_{r(r)} = 0, \quad (1.38)$$

$$U_{r(R)} = U_R.$$

Along the ~~axis~~ ^{tube} axis of ~~the~~

$$r = 0, U_r = 0, U_x = 0 \text{ with } X = 0.$$

Solution to the equations pointed out above with boundary conditions gives the following expression for the pressure differential in the vapor phase of the thermal ~~tube~~ ^{tube}:

$$\Delta P_n = P_{(0,r)} - P_{(x,r)} = 8\rho_n U_r^2 \left(\frac{X}{R} \right)^2 \left(\frac{1,325}{Re_r} + 0,617 \right). \quad (1.39)$$

If we suppose that in evaporator ~~the pressure~~ the pressure differential is equal ^{to} $\Delta P_{n(n)}$, ~~that then~~

$$\Delta P_{n(n)} = 8\rho_n U_r^2 \left(\frac{l_n}{R} \right)^2 \left(\frac{1,325}{Re_r} + 0,617 \right). \quad (1.40)$$

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The relation of the radial velocity of motion ~~to~~ ^{of vapor} to the axial

average speed ~~of~~ \bar{U}_x at output ~~of~~ from evaporator ~~can~~ can be obtained with the aid of the law of conservation of mass

$$\frac{U_r}{\bar{U}_x} = \frac{1}{2} \cdot \frac{R}{l_u} \quad (1.41)$$

analogously

$$\frac{Re_r}{Re_x} = \frac{1}{2} \cdot \frac{R}{l_u}, \quad (1.42)$$

$$Re_x = \frac{\bar{U}_x R}{\nu}, \quad Re_r = \frac{\bar{U}_r R}{\nu};$$

a) in evaporator ~~Re_r~~

$$\Delta P_{n(n)} = \left(1.234 + \frac{5.3}{Re_x} \cdot \frac{l_u}{R} \right) \rho_n \bar{U}_x^2 \quad (1.43)$$

for $Re \gg 1$ (laminar flow ^{of vapor} ~~is~~);

b) in the heat-insulated ^{to} zone the process of ~~the~~ hydrodynamic motion of vapor will be analogous to the process of the flow of gas in ~~the~~ ^{tube} with rough walls. It is possible to describe in the case of laminar flow by Poiseuille equation

$$\Delta P_{n(r)} = \frac{1}{2} \rho_n \bar{U}_x^2 \left(16 \frac{x}{R Re_x} \right); \quad (1.44)$$

c) in the zone of ~~the~~ condensation of ~~the~~ ^{tube} the process of condensation to a certain extent is analogous to the process of

suction through the porous ~~diaphragm~~ ^{wall}. In the presence of condensation ^{of vapor} the laminar wall boundary layer of condenser ~~capacitor~~ ^{is} the pressure differential ^{is} substantially less than in the same ~~diaphragm~~ ^{tube} but without condensation or suction; therefore it can be disregarded and considered that ⁱⁿ the condenser ~~pressure~~ ^{is} constant ^{and} is equal to pressure ^{of vapor} ~~at~~ at the inlet into condenser ~~capacitor~~.

Thus, a pressure drop in the vapor phase of ~~diaphragm~~ ^{tube} for the case of laminar flow ^{of vapor} ~~diaphragm~~ can be presented in the following form:

$$\begin{aligned} \Delta P_n &= \Delta P_{n(n)} + \Delta P_{n(\tau)} + \Delta P_{n(k)} = \\ &= \left(1,234 + \frac{5,3}{Re_x} \cdot \frac{l_n}{R} \right) \rho_n \bar{U}_x^2 + 0,5 \rho_n \bar{U}_x^2 \left(16 \frac{l_\tau}{R Re_x} \right). \quad (1.45) \end{aligned}$$

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In rough approximation for ~~the~~ long ^{tubes} ~~diaphragm~~ when $l_\tau \gg l_n$ and $Re_\tau \ll 1$,

a pressure drop both in the evaporator ~~evaporizer~~ and in condenser ~~condensizer~~ can be disregarded, then

$$\Delta P_n = \Delta P_{n(r)}. \quad (1.46)$$

If we assume that is valid the Poiseuille ^{law} ~~equation~~, then

$$\Delta P_n = -8 \frac{\mu l_{\text{eff}}}{\pi R^4} j_{nj}.$$

For the case of turbulent flow ^{of vapor} ~~in~~ in thermal ^{tube} ~~the~~ the pressure differential is determined as follows:

a). in evaporator ~~evaporizer~~ the pressure differential

$$\Delta P_{n(n)} = 4,45 \bar{U}_x^2 \frac{\rho_n U_{rn} l_n}{R}, \quad (1.47)$$

if one considers that $\frac{U_r}{U_x} = 0,5 \frac{R}{l_n}$, ~~then~~ then

$$\Delta P_{n(n)} = 2,23 \rho_n \bar{U}_x^2; \quad (1.48)$$

b) in the heat-insulated ^{te} part of the ~~tube~~ ^{tube}, according to the equation of Blasius,

$$\Delta P_{n(\tau)} = 0,0107 \frac{\mu_n^{1/4} l_\tau}{\rho_n R^{19/4}}; \quad (1.49)$$

c) in condenser ~~Aspirator~~

$$\Delta P_{n(k)} = 0. \quad (1.50)$$

Thus, ~~a common/general~~ total pressure differential in tube under the condition of turbulent flow ~~point~~ of vapor

$$\Delta P_n = 4,45 \rho_n \bar{U}_x^2 \frac{U_r}{R} l_n + 0,0107 \frac{\mu_n^{1/4} l_\tau}{\rho_n R^{19/4}}. \quad (1.51)$$

The total pressure differential in liquid and vapor phase. The total pressure differential in ^{vapor} ~~and~~ and liquid phase can be written in the form [111]

$$\Delta P_n \geq \left(\frac{\mu_n L_{\text{eq}}}{\rho_n K S} + \frac{8 \mu_n l_{a.3}}{\pi \rho_n R_n^3} \right) j + 0,075 \frac{j^2}{\rho_n R_n^2} + g \rho_n L \sin \theta. \quad (1.52)$$

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From this quadratic equation relative to flow j it is possible to find j and, therefore, to determine the heat output, transferred along the ~~axis of~~ ^{tube} ~~axis of~~ by formula

$$Q = j r'.$$

For ~~the~~ long thermal ^{tubes} ~~tubes~~ where the pressure differential is determined by viscous forces, expression for j takes form [107]

$$j = \frac{\Delta P_n - g \rho_n L \sin \theta}{L_{\text{eq}} \left[\frac{\mu_n}{\rho_n K S} + \frac{8 \mu_n}{\pi \rho_n R_n^3} \right]}. \quad (1.53)$$

Limitation on heat transfer in thermal ^{tubes} ~~tubes~~ as a result of the emergence of shock waves in vapor phase. If the speed of motion ^{of vapor} ~~in~~ in the condenser ~~resistor~~ ^{tube} of thermal ~~tubes~~ reaches the speed of sound, are formed the shock waves, which can cause the disturbance of ^{normal} ~~performance~~ ^{performance of tube.}

In connection with this it is necessary that the flow ^{of vapor} ~~in~~ ~~of~~ j_n would not exceed value [29]

$$j_{3n} = \pi r_n^2 U_{x.3n} \rho_n, \quad (1.54)$$

where ~~the~~ $U_{x.3n}$ - the speed of sound.

$U_{x, \text{th}}$ for a perfect gas can be found according to formula

$$U_{\text{ch. th}} = \sqrt{\frac{C_p p_{\text{th}}}{C_v \rho_{\text{th}}}} \quad (1.55)$$

The maximum heat output, transferred along thermal ~~output~~ ^{tubes} in this case is equal to

$$Q_{\text{th}} \leq j_{\text{th}} r' = \pi r_{\text{th}}^2 U_{x, \text{th}} \rho_{\text{th}} r' \quad (1.56)$$

Limitation on heat transfer in thermal ~~output~~ ^{tubes} as a result of the interaction of flow ~~output~~ ^{of vapor} with liquid in porous core. At high speeds of motion ~~output~~ ^{of vapor} it can substantially ~~slow down~~ the transfer of liquid on core. This first of all is related to thermal ~~output~~ ^{tubes} with open channels [62]. To evaluation criteria of interaction ~~output~~ ^{of vapor} with liquid are the criteria for Weber We [112]:

$$\text{We} = \frac{U^2 \rho_{\text{th}}}{\sigma} Z \leq 1, \quad (1.57)$$

where Z ~~output~~ is the significant dimension of the surface of the interaction of flow ~~output~~ ^{of vapor} with liquid.

The maximum heat output, transferred along the thermal ~~output~~ ^{tube} Q_{tp} and determined by interaction ~~output~~ ^{of vapor} with liquid, can be found from expression

$$Q_{\text{tp}} = U \rho_{\text{th}} r' = \sqrt{\frac{\rho_{\text{th}} \sigma r'^2}{Z}} \quad (1.58)$$

with $\text{We} = 1$.

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2. Thermal ~~data~~ ^{tubes} of ~~the~~ moderate temperature range and low-temperature ~~data~~ ^{tubes}.

^{influenced by} The transmitting of heat along low-temperature thermal ~~data~~ ^{tubes is} ~~affected~~ a series of ~~the~~ parameters, such, as thermal resistance of the walls of ~~the~~ ^{tube} porous core, saturated by liquid, temperature jump in the zone of evaporation and condensation, the thermal resistance of fluid film above the porous core in the zone of condensation and finally the transport properties of porous core.

All parameters pointed out above, with the exception of the

latter, characterize ^{the} process heat- and mass exchange within thermal ^{the} ~~tube~~ when the gradient of ~~the~~ temperature is present, as motive power. The motive power of the transfer of liquid in capillary-porous body is the gradient of capillary potential, which is formed in the presence of evaporation and condensation in the different parts of the porous body. For low-temperature thermal ^{tubes} ~~ducts~~ are characteristic boundary conditions ^{3, 2 or 1} kind in the zone of condensation.

The maximum value of ~~the~~ heat output ^{Q_{max}} ~~of ducts~~ transferred along ^{tube} ~~duct~~ is limited, on one hand, by the emergence of the crisis of boiling ^{of} liquid in the pores of core, ^{on the} ~~with~~ other ^{by} - limited capacity during the transfer of liquid on porous core under the action of the gradient of capillary potential. Thermal ducts can work either in the mode of the evaporation of liquid from the surface of porous body or in the mode ~~conditions~~ of boiling.

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Under conditions of weightlessness with heat removal by the evaporation of liquid from the surface of ^a ~~the~~ flat ~~porous~~ porous core ^{Q_{max}} it is possible to find ~~from~~ formula

$$Q_{\max} = \frac{2\sigma}{R_{\min}} K \frac{\rho_{\text{ж}} r'}{\mu_{\text{ж}}} \cdot \frac{S}{\left(\frac{L_{\text{ж}}}{2} + L_{\text{а}} + \frac{L_{\text{к}}}{2}\right)}, \quad (1.59)$$

~~that~~ which was obtained on the basis of the fact that is observed the equality

$$\Delta P_{\text{ж}} = \Delta P_{\text{жк}} + \Delta P_{\text{м}},$$

in this case we assume that $\Delta P_{\text{м}} \approx 0$.

the value of Q_{\max} during the emergence of the crisis of boiling usually is ~~found~~ ^{found} experimentally. Besides knowledge of Q_{\max} for low-temperature ~~tubes~~ ^{tubes}, it is necessary to know their thermal resistance or the temperature differential between the external surface of evaporator ~~capacitor~~ and condenser ~~capacitor~~ at the known value Q .

Let us examine process ^{of} heat- and mass exchange in the

↑

evaporator, ~~vapORIZER~~ ^{tube} of thermal ~~core~~ with flat, ~~plate~~ porous core (Fig. 4). Let us assume that the porous core has a thickness δ and width b , is isotropic and the heat exchange in it is realized by thermal conductivity and convection.

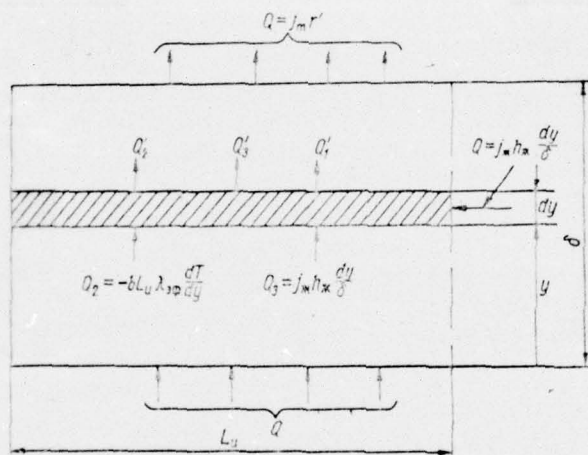


Fig. 4.

Fig. 4. ~~Cell~~^a element of ~~the~~ flat ~~plate~~ porous core of thermal tube in the zone of evaporation.

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The properties of liquid and core are determined at the averaged from the volume of core temperature. Temperature jumps in the zone of evaporation and condensation we disregard. Fluid film above the porous core in the zone of condensation is absent.

For the ~~cell~~ element of flat ~~plate~~ porous core with a thickness δ in the zone of ~~the~~ evaporation of thermal ~~plate~~^{tube}, depicted ~~on~~ⁱⁿ Fig. 4, it is possible to write the following equations of heat balance:

<p>(1) Вход</p> $Q_1 + Q_3 = j_m h_m \frac{y}{\delta} + j_m h_m \frac{dy}{\delta},$ $Q_2 = -b L_n \lambda_{np} \frac{dT}{dy},$ $Q_1 + Q_2 + Q_3 = Q_1 + Q_2 + Q_3 + dQ_1 + dQ_2 + dQ_3,$ $dQ_1 + dQ_2 + dQ_3 = 0.$	<p>(2) Выход</p> $Q'_2 + Q'_3 + Q'_1 = Q_1 + Q_2 + Q_3 + dQ_1 + dQ_2 + dQ_3 \quad (1.60)$
--	---

Key (1) Inlet; (2) Outlet.

On the basis of energy balance in the ~~cell~~ element of porous core, let us write

$$-bL_n\lambda_{n\phi}\frac{d}{dy}\left(\frac{dT}{dy}\right)dy + j_n\frac{dh}{dy} \cdot \frac{ydy}{\delta} = 0. \quad (1.61)$$

The derivatives of the second order in this case we disregard in ~~view~~
~~view~~ of their smallness

$$-bL_n\lambda_{n\phi}\frac{d^2T}{dy^2}dy + j_n\frac{y}{\delta} \cdot \frac{dh_n}{dy}dy = 0, \quad (1.62)$$

$$\frac{d^2T}{dy^2} = \frac{j_n}{bL_n\lambda_{n\phi}\delta}y \frac{dh_n}{dy}. \quad (1.63)$$

Let us make the following substitution:

$$\frac{dh_n}{dy} = C_{P_n}\frac{dT}{dy} + \frac{1}{\rho_n}\frac{dP_n}{dy}, \quad (1.64)$$

where ~~the~~ $C_{p_{\kappa}}$ - the heat capacity of the porous core, filled by liquid.

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In the cores of thermal ^{tubes} ~~cores~~, the term ~~of~~ dP_m/dy usually composes ^a ~~the~~ very low value, ~~so~~ which it is possible to disregard, then

$$\frac{dh_{\kappa}}{dy} \approx C_{p_{\kappa}} \frac{dT}{dy} \quad (1.65)$$

substituting (1.65) in (1.62), we obtain

$$\frac{d^2 T}{dy^2} = \frac{j_{\text{ж}} C_{P_{\text{ж}}}}{b L_{\text{н}} \delta \lambda_{\text{эф}}} y \frac{dT}{dy}. \quad (1.66)$$

Let us designate

$$\frac{dT}{dy} = z,$$

then

$$\frac{dz}{dy} = \frac{j_{\text{ж}} C_{P_{\text{ж}}}}{b L_{\text{н}} \delta \lambda_{\text{эф}}} y z, \quad (1.67)$$

$$\frac{dz}{z} = \frac{j_{\text{ж}} C_{P_{\text{ж}}}}{b L_{\text{н}} \delta \lambda_{\text{эф}}} y dy, \quad (1.68)$$

$$\ln C_1 z = \frac{j_{\text{ж}} C_{P_{\text{ж}}}}{b L_{\text{н}} \delta \lambda_{\text{эф}}} y^2, \quad (1.69)$$

$$z = \frac{1}{C_1} \exp \left(\frac{j_{\text{ж}} C_{P_{\text{ж}}}}{b L_{\text{н}} \delta \lambda_{\text{эф}}} y^2 \right). \quad (1.70)$$

Let us write the boundary conditions: with $y = 0$

$$-bL_n\lambda_{\varphi\Phi} \frac{dT}{dy} = Q, \quad (1.71)$$

$$\frac{1}{C_1} = -\frac{Q}{bL_n\delta\lambda_{\varphi\Phi}}, \quad C_1 = -\frac{bL_n\delta\lambda_{\varphi\Phi}}{Q}, \quad (1.72)$$

$$\frac{dT}{dy} = -\frac{Q}{bL_n\delta\lambda_{\varphi\Phi}} \exp\left(\frac{j_{\kappa}C_P\kappa}{2bL_n\delta\lambda_{\varphi\Phi}} y^2\right); \quad (1.73)$$

with $y = \delta$

$$T = T_{\text{nac}}, \quad (1.74)$$

$$\int_{T_i}^{T_{\text{nac}}} dT = -\frac{Q}{bL_n\delta\lambda_{\varphi\Phi}} \int_{y_i}^{\delta} \exp\left(\frac{j_{\kappa}C_P\kappa}{2bL_n\delta\lambda_{\varphi\Phi}} y^2\right) dy. \quad (1.75)$$

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Dual integration of differential equation (1.63) gives to us temperature field in the porous core, filled by heat-transfer agent, when the evaporation occurs from the surface of porous core.

After dual integration we obtain

$$T_1 - T_{\text{nac}} = - \frac{Q}{bL_n \delta \lambda_{\text{эф}}} \int_{y_1}^{\delta} \exp \left(\frac{j_{\text{ж}} C_{p_{\text{ж}}}}{2bL_n \delta \lambda_{\text{эф}}} y^2 \right) dy. \quad (1.76)$$

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The temperature differential between the external and internal surfaces of the core of the evaporator ~~is equal to~~ of thermal ^{tube} ~~conductance~~ is equal to 1

$$T_n - T_{nac} = -\frac{Q\delta}{bL_n\lambda_{\text{эф}}''} \int_0^y \exp\left(\frac{j_n C_{p,ж} y^2}{2bL_n\lambda_{\text{эф}}''}\right) dy. \quad (1.77)$$

Analogously ~~is~~ ^{found} ~~is~~ the temperature differential between the external and internal surfaces of the condenser ~~capacitor~~ of thermal ~~tube~~ ^{tube}.

$$T_{\text{nac}} - T_{\text{K}} = \frac{Q\delta}{bL_{\text{K}}\lambda_{\text{g}\phi}^{\text{K}}} \int_0^y \exp\left(\frac{-j_{\text{K}}y^2 C_{\text{P}\text{K}}}{2bL_{\text{K}}\lambda_{\text{g}\phi}^{\text{K}}}\right) dy. \quad (1.78)$$

The temperature differential on the wall of the housing of evaporator ~~capacitor~~ and condenser ~~capacitor~~ is equal to

$$\Delta T_{\text{H}} = \frac{Q}{bL_{\text{H}}\lambda_{\text{et}}} \delta_{\text{L}}, \quad \Delta T_{\text{K}} = \frac{Q}{bL_{\text{K}}\lambda_{\text{et}}} \delta_{\text{L}}. \quad (1.79)$$

Total temperature differential between the external wall of evaporator ~~boiler~~ and condenser ~~boiler~~ of flat ~~plane~~ low-temperature thermal ~~unit~~ ^{tube} or steam chamber is equal to

$$T_{\text{нар}}^{\text{н}} - T_{\text{нар}}^{\text{к}} = \Delta T_{\text{н}} + (T_{\text{н}} - T_{\text{нас}}) + (T_{\text{нас}} - T_{\text{к}}) + \Delta T_{\text{к}} =$$

$$= \frac{Q\delta_1}{bL_{\text{н}}\lambda_{\text{ст}}} + \frac{Q\delta}{bL_{\text{н}}\lambda_{\text{сф}}^{\text{н}}} \int_0^{\delta} \exp\left(\frac{j_{\text{н}}y^2 C_{P_{\text{ж}}}}{2bL_{\text{н}}\lambda_{\text{сф}}^{\text{н}}}\right) dy +$$

$$+ \frac{Q\delta}{bL_{\text{к}}\lambda_{\text{сф}}^{\text{к}}} \int_0^{\delta} \exp\left(\frac{-j_{\text{к}}y^2 C_{P_{\text{ж}}}}{2bL_{\text{к}}\lambda_{\text{сф}}^{\text{к}}}\right) dy + \frac{Q\delta_1}{bL_{\text{к}}\lambda_{\text{ст}}}, \quad (1.80)$$

$$\int_0^{\delta} \exp(Ay^2) dy = \int_0^{\delta} \exp\left(\frac{-j_{\text{н}}C_{P_{\text{ж}}}}{2bL_{\text{н}}\lambda_{\text{сф}}^{\text{н}}} y^2\right) dy.$$

When
~~at~~ $y < 1$
0

$$A = \frac{j_{\text{н}} C_{p_{\text{ж}}}}{2bL\lambda_{\text{эф}}}$$

$$\int_0^{\delta} \exp(Ay^2) dy = \delta + \frac{A\delta^3}{3} + \frac{A^2\delta^5}{2!5} + \frac{A^3\delta^7}{3!7} + \dots$$

In cylindrical thermal ~~anal~~^{tube} analogous analysis ~~re~~ gives

$$T_{\text{н}} - T_{\text{нас}} = \frac{Q}{2\pi r_{\text{нар}}^{\phi} L_{\text{н}} \delta \lambda_{\text{эф}}^{\text{н}}} \int_0^{\delta} \exp\left(\frac{j_{\text{н}} C_{p_{\text{ж}}} y^2}{4\pi r_{\text{нар}}^{\phi} L_{\text{н}} \delta \lambda_{\text{эф}}^{\text{н}}}\right) dy, \quad (1.81)$$

$$T_{\text{нас}} - T_{\text{к}} = \frac{Q}{2\pi r_{\text{нар}}^{\phi} L_{\text{к}} \delta \lambda_{\text{эф}}^{\text{к}}} \int_0^{\delta} \exp\left(\frac{-j_{\text{к}} C_{p_{\text{ж}}} y^2}{4\pi r_{\text{нар}}^{\phi} L_{\text{к}} \delta \lambda_{\text{эф}}^{\text{к}}}\right) dy, \quad (1.82)$$

In the presence on external surface ^{of} the evaporator ~~the porous~~ and the condenser ~~the porous~~ of the low-temperature thermal ^{tubes} ~~tubes~~ of the boundary conditions of the 1st kind ~~of~~ ($T_{\text{nap}}^{\text{v}} = \text{const}$; $T_{\text{nap}}^{\text{c}} = \text{const}$) thermal power Q , transferred along thermal duct, can be determined by formula

$$Q = (T_{\text{nap}}^{\text{v}} - T_{\text{nap}}^{\text{c}}) b \left[\frac{\delta_1}{L_{\text{R}} \lambda_{\text{er}}} + \frac{\delta}{L_{\text{R}} \lambda_{\text{ph}}^{\text{v}}} \times \right. \\ \times \int_0^{\delta} \exp \left(\frac{j_{\text{R}} y^2 C_{\text{p}}^{\text{v}}}{2b L_{\text{R}} \lambda_{\text{ph}}^{\text{v}}} \right) dy + \frac{\delta}{L_{\text{R}} \lambda_{\text{ph}}^{\text{c}}} \times \\ \left. \times \int_0^{\delta} \exp \left(\frac{-j_{\text{R}} y^2 C_{\text{p}}^{\text{c}}}{2b L_{\text{R}} \lambda_{\text{ph}}^{\text{c}}} \right) dy + \frac{\delta_1}{L_{\text{R}} \lambda_{\text{er}}} \right]^{-1} \leq Q_{\text{max}}. \quad (1.83)$$

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This analysis is valid when in the porous core of thermal ^{tube} ~~tube~~ is absent the free convection of liquid, caused by the gradient of temperature field in the radial direction of core, i.e., the

criterion for $Ra_{\text{eff}} < 4\pi^2$ [97]:

$$Ra_{\text{eff}} = g \left(\frac{\alpha' C_m \rho_m}{\nu} \right) \left(\frac{K}{\lambda_{\text{eff}}} \right) \Delta T. \quad (1.84)$$

In this case, the speed of the filtration motion of liquid in the axial direction of core exceeds the speed of the displacement of liquid under the action of the forces of ~~the~~ free convection, caused by the presence of density gradient as a result of the existence of the gradient of the temperature in the cross section of core.

3. Effect of the boundary conditions in the zone of condensation on the value of heat flux, the distribution of temperature field and the thermal resistance of ~~the~~ thermal ^{tubes} ~~cores~~ and steam chambers.

The boundary conditions in the zone of condensation ^{of vapor on} ~~the pair to~~ ^a cooled porous surface, and the also boundary conditions on the external wall of the condenser ~~resistor~~ ^{pipes} of the thermal ~~cores~~ and steam chambers determine the fundamental performance characteristics of thermal ^{tubes} ~~cores~~. Specifically, on them depends the value of the

temperature of ~~the~~ saturation ~~of~~ T_{vac} , the thermal resistance of ~~the~~ ^{tube} as a whole, the temperature differential along the external wall of the ~~the~~ ^{tube} or steam chamber. In this paragraph let us attempt to examine the effect of the boundary conditions of the 2nd kind on the work of thermal ~~the~~ ^{tubes} with flat ~~plane~~ and cylindrical porous core depending on the characteristics of capillary-porous bodies and liquids in the presence and absence of fluid film above the surface of condensation.

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This analysis makes it possible to determine the most successful combination of porous core and heat-transfer agent with the assigned heat flux on the external surface of the condenser of thermal ~~the~~ ^{tube}, and also the necessary length of the condenser ~~capacitor~~ of thermal ~~the~~ ^{tube}. When conducting this analysis, is made a series of ~~the~~ significant assumptions, basic ~~the~~ ^{of} which ~~is~~ ^{is} the absence of the gradient of ~~the~~ temperature in the cross section of porous core.

One-dimensional model

1. Dependence of the geometric dimensions of ~~the~~ ^a flat ~~plane~~

porous core of the condenser ~~capacitor~~ of the steam chambers and thermal ^{tubes} ~~losses~~ on the amount of heat, scattered on wall into the environment (boundary conditions of the 2nd kind, $q_w = \text{const}$).

Let us make the following assumptions:

in condenser ^{the} ~~capacitor~~ liquid in the pores of core ^{on} ~~at~~ entire length has constant temperature, i.e., there is no ~~is~~ supercooling; heat exchange with the wall of housing takes ^{place with} ~~a course of~~ forced convection;

heat flux on the external surface of condenser is constant, $q_n = Q/L_n$, $C = \text{const}$ ^{and} evenly is scattered into the environment;

saturated ^{vapor} ~~steam~~ is condensed on the surface of core directly in pores (there is no fluid film on the surface of porous body) at the constant velocity $U_n = \text{const}$; flow ^{of vapor} ~~the vapor~~ does not introduce ^a ~~the~~ contribution to a change in the momentum of liquid in porous core;

the interface liquid - vapor in pores is characterized by radius of curvature R ;

fluid flow in porous core ^{is} ~~is~~ laminar, obeys the law of Darcys and has a velocity of U_m .

the porous core of condenser ~~capacitor~~ is examined in the form of parallelepiped. The ~~element~~ element of condenser ~~capacitor~~ (Fig. 5a) has geometric dimensions (dx, b, c) and a porosity of ~~mass~~ Π .

let us examine the integral equations of mass balance and energy for this ~~element~~ element.

Balance of mass. Let us find the dependence between the flow ^{of} ~~vapor~~ and liquids on the basis of the equation of continuity.

~~Figure 5a.~~

For the ~~element~~ ^{and} element of porous core ^{by} width $z = b$, ~~with~~ height ~~by~~ $y = c$ and with a thickness dx fluid flows at entrance and exit are equal ~~to~~ respectively ^{to}

$$j_{\text{H}(1)} + j_{\text{H}} = j_{\text{H}(2)}, \quad (1.85)$$

$$j_{\text{H}(2)} = \rho_{\text{H}} \Pi(bc) U_{\text{H}}, \quad (1.86)$$

$$U_{\text{H}(2)} = U_{\text{H}(1)} + dU_{\text{H}}, \quad (1.87)$$

$$j_{\text{H}(2)} = \rho_{\text{H}} \Pi(bc) U_{\text{H}(2)} = \rho_{\text{H}} \Pi(bc) [U_{\text{H}(1)} + dU_{\text{H}}], \quad (1.88)$$

$$j_{\text{H}} = \rho_{\text{H}} \Pi(bc) \frac{dU_{\text{H}}}{dx} dx = \rho_{\text{H}} U_{\text{H}} (b dx). \quad (1.89)$$

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During the motion of liquid along porous core under the action of a pressure difference as a result of the presence of the gradient of capillary potential it is necessary to overcome the forces of friction and inertia. However, in the majority of cases inertia terms

can be disregarded. The viscous ^{resistance} ~~drag~~ of porous core changes the momentum of liquid. Under stationary conditions

$$\begin{aligned}(P_1 - P_2) A - F_{\text{внтр}} &= \rho U_{(1)}^2 \Pi(bc) - \rho U_{(2)}^2 \Pi(bc) = \\ &= \rho \frac{d(U^2)}{dx} dx \Pi(bc).\end{aligned}\quad (1.90)$$

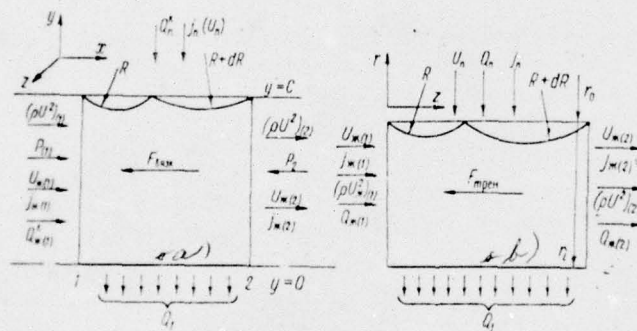


Fig. 5. Element of the condenser, a) in the Cartesian coordinate system; b) in cylindrical coordinate system.

According to the law of Darcy, during the ~~viscous~~ ^{laminar} motion of liquid in porous body

$$\frac{dP}{dx} = K_1 \frac{\mu_m}{\rho_m} \cdot \frac{j_m}{A}, \quad (1.91)$$

where A - the cross-sectional area of porous core,

$$F_{\text{вязк}} = \frac{dP}{dx} dx \Pi(bc) = K_1 \Pi^2(bc) \mu_m U_m dx. \quad (1.92)$$

As a result we obtain the differential equation of a change in the momentum during flow of liquid through the ~~element~~ element of porous core

$$-2\sigma \frac{dR}{R^2} - \Pi K_1 \mu_m U_m dx = \rho_m \frac{d(U_m^2)}{dx} dx. \quad (1.93)$$

Energy balance on the basis of the law of conservation of energy. Let us find the dependence between the heat flux, transferred to wall $b dx$ of the ~~element~~ element of the porous core dx by the convection current of liquid, and by the heat flux, isolated during condensation of ~~vapor~~ vapor on the surface of the ~~element~~ element of the porous core:

$$Q_n^k + Q_{n(1)}^k = Q_{n(2)}^k + Q_1, \quad (1.94)$$

$$Q_n^k = j_n h_n, \text{ где}$$

$$j_n = \rho_m \frac{dU_m}{dx} dx \Pi(bc), \quad (1.95)$$

$$Q_1 = q(b dx), \quad (1.96)$$

$$Q_{n(2)}^k = j_m h_m + \frac{d(j_m h_m)}{dx} dx, \quad (1.97)$$

where

$$i_m = \rho_m \Pi (cb) U_m. \quad (1.98)$$

Consequently,

$$U_m \rho_m \frac{dh_m}{dx} - (h_n - h_m) \rho_m \frac{dU_m}{dx} + \frac{q_n}{c \Pi} = 0, \quad (1.99)$$

but $dh_m/dx \approx 0$, since ~~are~~ ^{over the} assumed ~~to be~~ the condition of isothermal flow of liquid ~~in~~ core, therefore,

$$\frac{dU_m}{dx} = \frac{q_n}{h_{nm} c \Pi \rho_m}. \quad (1.100)$$

If this expression is integrated over x , then we will obtain

$$U_m = \frac{q_n}{h_{nm} c \Pi \rho_m} x. \quad (1.101)$$

Respectively fluid flow on porous core in condenser is equal to

$$i_m = \rho_m \Pi (bc) U_m = \frac{q_n b}{r'} x. \quad (1.102)$$

If we combine the equation of mass balance and energy, then we will obtain the integral equation of energy transfer and substance in the condenser ~~capacitor~~ of thermal ~~pipe~~ pipe.

$$\begin{aligned}
 - \int_{R_X=0}^{R_{X=L_K}} 2\sigma \frac{dR}{R^2} - \int_0^{L_K} K_1 \frac{q_R}{r'C} \frac{\mu_R}{\rho_R} x dx = \\
 = \int_0^{L_K} \frac{2q_R^2 x dx}{r'^2 \rho_R \Pi^2 C^2}, \quad (1.103)
 \end{aligned}$$

where ~~the~~ L_K - the length of condenser ~~capacitor~~.

The maximum capacity of the porous core of condenser ~~capacitor~~ can be evaluated, if we determine flow or velocity of liquid at the maximum length of the condenser ~~capacitor~~ of L_{Kmax} . In this case, it is necessary to know a radius of interface liquid - ~~pipe~~ vapor with $x = 0$

and with $x = L_{\kappa \max}$, i.e. to evaluate the capillary pressure head, created by the field gradient of capillary forces along the porous core of condenser ~~capacitor~~ ^{with} by the length ~~of~~ $L_{\kappa \max}$.

With $x = 0$ it is possible to assume that the radius of curvature of interface (liquid - ~~gas~~ ^{vapor}) approaches infinity, since the condensation occurs on the surface of the porous body ~~at~~ $R_{x=0} \rightarrow \infty$.

The minimum value of R_{\min} ^{$x=L_{\kappa}$} can be estimated from experiments regarding the maximum ~~height~~ ^{capillary} lifting of liquid in porous core against the force of gravitation

$$R_{\min} = \frac{2\sigma}{\rho g h_{\max}} \quad (1.104)$$

By knowing integration limits for $R(R_{x=0}, R_{x=L_{\kappa}})$, equation (1.103) it is possible to solve relative to L_{κ}

$$L_{\kappa \max} = \left(\frac{\rho_{\kappa} r' \sigma}{\mu_{\kappa}} \right)^{1/2} \times \left(\frac{4c}{q_{\kappa} R_{\min} K_1 \left(1 + \frac{2q_{\kappa}}{\mu_{\kappa} r' C \Pi^2 K_1} \right)} \right)^{1/2} \quad (1.105)$$

In the majority of cases this expression it is possible to simplify to

$$L_{\kappa \max} = 2 \sqrt{\frac{\rho_{\kappa} r' \sigma c}{\mu_{\kappa} q_{\kappa} K_1 R_{\min}}} \quad (1.106)$$

since value

$$\frac{2q_{\text{н}}}{\mu_{\text{н}} r' C \Pi^2 K_1} \ll 1.$$

The heat transfer rate, removed from the surface of condenser ~~condenser~~ with its assigned length and ~~the~~ thickness of core, is determined from formula

$$q_{\text{н}} = -K_1 \mu_{\text{н}} r' \Pi^2 C - \left(K_1^2 \mu_{\text{н}}^2 r' \Pi^4 C^2 + \frac{4r' \rho_{\text{н}} \Pi^2 C^2 \sigma}{L_{\text{н}}^2 R_{\text{min}}} \right)^{1/2}. \quad (1.107)$$

The total amount of heat, scattered by condenser ~~apparatus~~, is equal to

$$Q_n = \int_0^{L_n} q_n b dx = q_n b L_n = -b L_n \left[K_{1n} u_{nR} r' \Pi^2 C - \left(K_{1n}^2 u_{nR}^2 r'^2 \Pi^4 C^2 + \frac{4 r'^2 \rho_{nR} \Pi^2 C^2 \sigma}{L_n^2 R_{min}} \right)^{1/2} \right]. \quad (1.108)$$

2. Dependence of the geometric dimensions of the porous core of the condenser ~~apparatus~~ of thermal ^{tubes} ~~ducts~~ in the form of hollow cylinder on the amount of heat, scattered on wall into the environment (boundary conditions of the 2-kind $q_n = \text{const}$).

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Let us make a series of ~~the~~ assumptions:

the temperature of liquid in the porous core of condenser ~~capillary~~ is constant; heat exchange with the wall of thermal ~~heat~~ ^{tube} takes ^{place by means} ~~a course~~ of forced convection;

Heat flow on external surface is constant

$$q_R = \frac{\lambda(t_1 - t_2)}{r \ln \left(\frac{r_2}{r_1} \right)}, \quad q_R = \frac{Q}{2\pi r L_R} \quad (1.109)$$

and evenly is scattered in the environment (Fig. 5b);

saturated ^{vapor} ~~steam~~ is condensed on the internal surface of the porous core of condenser ~~capillary~~ (fluid film is absent) with constant speed of $U_n = \text{const}$;

the interface liquid - vapor in pores is characterized by radius of curvature R;

the motion of liquid in porous core obeys the law of **Darcy**;

flow ^{of vapor} ~~does not~~ introduce ^a ~~contribution~~ to a change in the momentum of liquid in porous core.

Balance of mass

$$j_{\text{ж}(1)} + j_{\text{n}} = j_{\text{ж}(2)}, \quad (1.110)$$

$$j_{\text{ж}(1)} = \rho_{\text{ж}} \Pi (\pi r_l^2 - \pi r_0^2) U_{\text{ж}}, \quad (1.111)$$

$$U_{\text{ж}(2)} = U_{\text{ж}(1)} + dU_{\text{ж}}, \quad (1.112)$$

$$j_{\text{ж}(2)} = \rho_{\text{ж}} \Pi (\pi r_l^2 - \pi r_0^2) U_{\text{ж}(2)} = \\ = \rho_{\text{ж}} \Pi (\pi r_l^2 - \pi r_0^2) [U_{\text{ж}(1)} + dU_{\text{ж}}], \quad (1.113)$$

$$j_{\text{n}} = \rho_{\text{n}} U_{\text{n}} \Pi 2\pi r_0 dz = j_{\text{ж}(2)} - j_{\text{ж}(1)} = \\ = \rho_{\text{ж}} \Pi (\pi r_l^2 - \pi r_0^2) \frac{dU_{\text{ж}}}{dz} dz, \quad (1.114)$$

$$F_{P_1} - F_{P_2} - F_{\text{тр}} = \rho U_1^2 \Pi (\pi r_l^2 - \pi r_0^2) - \\ - \rho U_2^2 \Pi (\pi r_l^2 - \pi r_0^2) = \rho \frac{d(U^2)}{dz} dz \Pi (\pi r_l^2 - \pi r_0^2). \quad (1.115)$$

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According to the law of Darcy,

$$F_{\text{тпен}} = \frac{dP}{dz} dz \cdot \Pi (\pi r_1^2 - \pi r_0^2) = K_1 \frac{\mu_m}{\rho_m} j_m \Pi dz \quad (1.116)$$

or

$$F_{\text{тпен}} = K_1 \frac{\mu_m}{\rho_m} \Pi (\pi r_1^2 - \pi r_0^2) U_m \Pi dz =$$

$$= K_1 \Pi^2 (\pi r_1^2 - \pi r_0^2) \mu_m U_m dz, \quad (1.117)$$

$$F_{P_i} = \left(P_n - \frac{2\sigma}{R} \right) \Pi \pi (r_1^2 - r_0^2), \quad (1.118)$$

$$F_{P_i} = \left(P_n - \frac{2\sigma}{R + dR} \right) \Pi \pi (r_1^2 - r_0^2), \quad (1.119)$$

$$F_{P_i} - F_{P_i} = -\Pi \pi (r_1^2 - r_0^2) \left(\frac{2\sigma}{R} - \frac{2\sigma}{R + dR} \right) =$$

$$= \rho_m \frac{d(U_m^2)}{dr} dr, \quad (1.120)$$

$$2\sigma \frac{dR}{R^2} - \Pi K_1 \mu_m U_m dz = \rho_m \frac{d(U_m^2)}{dr} dr. \quad (1.121)$$

energy balance

$$Q_{m(1)} + Q_n = Q_{m(2)} + Q_1, \quad (1.122)$$

$$Q_{m(1)} = j_m h_m = h_m \rho_m \Pi (\pi r_i^2 - \pi r_0^2) U_{m1}, \quad (1.123)$$

$$Q_n = j_n h_n = h_n \rho_m \Pi (\pi r_i^2 - \pi r_0^2) \frac{dU_m}{dr} dz, \quad (1.124)$$

$$\begin{aligned} Q_{m(2)} &= j_m h_m + \frac{d(j_m h_m)}{dz} dz = \\ &= h_m \rho_m \Pi (\pi r_i^2 - \pi r_0^2) \left(U_m + \frac{dU_m}{dz} dz \right), \end{aligned} \quad (1.125)$$

$$Q_1 = q_n 2\pi r_i dz. \quad (1.126)$$

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Summed up, we will obtain

$$\begin{aligned} (h_n - h_m) \rho_m \Pi (\pi r_i^2 - \pi r_0^2) \frac{dU_m}{dz} = \\ = \rho_m \Pi (\pi r_i^2 - \pi r_0^2) U_m \frac{dh_m}{dz} + q_n 2\pi r_i, \end{aligned} \quad (1.127)$$

$$\begin{aligned} U_m \frac{dh_m}{dz} \rho_m \Pi (\pi r_i^2 - \pi r_0^2) - (h_n - h_m) \rho_m \Pi (\pi r_i^2 - \pi r_0^2) \times \\ \times \frac{dU_m}{dz} + q_n 2\pi r_i = 0, \end{aligned} \quad (1.128)$$

but

$$\frac{dh_m}{dz} = 0,$$

$$r' \rho_m \Pi (\pi r_i^2 - \pi r_0^2) \frac{dU_m}{dz} = q_n 2\pi r_i, \quad (1.129)$$

$$\frac{dU_m}{dz} = \frac{q_n 2\pi r_i}{r' \rho_m \Pi (\pi r_i^2 - \pi r_0^2)}, \quad (1.130)$$

$$U_m = \frac{2q_n \pi r_i}{r' \rho_m \Pi (\pi r_i^2 - \pi r_0^2)} z, \quad (1.131)$$

$$j_m = \rho_m \Pi (\pi r_i^2 - \pi r_0^2) U_m = \frac{2q_n \pi r_i}{r'} z. \quad (1.132)$$

The ~~general~~ integral equation of energy transfer, substance and momentum ~~is~~ takes the form

$$\begin{aligned}
 - \int_{R_z=0}^{R_z=L_h} 2\sigma \frac{dR}{R^2} - \int_0^{L_h} K_{1111} \frac{2q_n \pi r_i}{r'^2 \rho_n \Pi^2 (\pi r_i^2 - \pi r_0^2)^2} dz = \\
 = \int_0^{L_h} \frac{4q_n^2 \pi^2 r_i^2}{r'^2 \rho_n \Pi^2 (\pi r_i^2 - \pi r_0^2)^2} dz. \quad (1.133)
 \end{aligned}$$

~~Eq. 1.133.~~

If we substitute the limit of ~~the~~ integration ~~of~~ R_{min} and $R = -$,

then

$$\frac{2\sigma}{R_{\min}} - K_1 \frac{\mu_K}{\rho_K} \cdot \frac{2q_K r_i}{r' (r_i^2 - r_0^2)} \times$$

$$\times \frac{L_K^2}{2} = \frac{4q_K^2 r_i^2 \pi^2}{r'^2 \rho_K \Pi^2 (r_i^2 - r_0^2)} \frac{L_K^2}{2}, \quad (1.134)$$

$$q_K = -K \left(\frac{K^2}{2} - \frac{r'^2 \rho_K \Pi^2 (r_i^2 - r_0^2)}{r_i^2 L_K^2} \frac{\sigma}{R_{\min}} \right)^{1/2}, \quad (1.135)$$

where

$$K = \frac{r' \mu_K K_1 \Pi^2}{4\pi r_i},$$

$$Q_{\max} = \int_0^{L_K} q_K 2\pi r_i dz = q_K 2\pi r_i L_K =$$

$$= -m - \left(\frac{m^2}{2} - 4\pi^2 \Pi^2 (r_i^2 - r_0^2) \frac{\sigma}{R_{\min}} \right)^{1/2}, \quad (1.136)$$

$$m = r' K_1 \Pi^2 \frac{L_K}{2},$$

$$L_K = \left[\frac{\frac{\sigma}{R_{\min}}}{\frac{\Pi^2}{\rho_K \Pi^2} + K_1 \frac{\mu_K}{\rho_K} \frac{\Pi}{2}} \right]^{1/2}, \quad (1.137)$$

$$\Pi = \frac{q_K r_i}{r' (r_i^2 - r_0^2)}.$$

Thus, the value of heat transfer rate ^{on a} ~~to the~~ unit of the external surface of the condenser ~~capacities of the~~ q_{kr} of the property ^{of} liquid and porous core, and also the geometric dimensions of the latter determine the necessary length of condenser ~~capacitor~~ with the assigned cross section, ^{and} ~~but also~~ also the maximum amount of heat ~~of~~ Q_{max} transferred ^{by} ~~according to~~ the thermal ^{tubes} ~~and~~ or the steam chamber. In this case, it is assumed that the heat removal in the zone of evaporator ~~capacities~~ is realized by the evaporation of liquid from pores near the surface of core (process of boiling liquid is absent).

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When conducting this analysis, it was assumed that the thermal resistance of porous core in the zone of condensation ^{is} negligibly ~~little~~ and temperature gradient in the cross section of core can be disregarded. This assumption is correct when using in the thermal ^{tubes} ~~ducts~~ and the steam chambers of ~~the~~ fine ~~thin~~ porous cores, which have high thermal conductivity (for example in the form of 2-3 layers of copper net or thin layer of ~~the~~ sintered copper shaving), and the liquids, which have high thermal conductivity (liquid sodium, water), and also when ~~the~~ small specific fluxes q_{kr} q_{kr} are present.

An example of the calculation of condenser ~~capacities~~ in the form

of hollow cylinder. The assigned parameters: liquid ~~is~~ ethyl alcohol; core - grid made of ~~the~~ stainless steel ^{with} cell 0, 15 x ^{0.15} ~~mm~~ ^{hmm}
(Size ~~dimension 0.45 mm~~; $R_{min} = 0,3 \text{ mm}$; $r_i = 20 \text{ mm}$; $r_0 = 19 \text{ mm}$; $r' =$
 $1.112 \cdot 10^{10} \text{ erg/g}$; $\sigma = 18.3 \text{ erg/cm}^2$ with $T = 70^\circ\text{C}$; $\rho_{\text{ж}} = 0.79 \text{ g/cm}^3$;
 $\Pi = 0.7$; $\kappa_1 = 6 \cdot 10^{-5} \text{ cm}^2$; $\mu_{\text{ж}} = 0,5 \cdot 10^{-2} \text{ g/cm} \cdot \text{s}$; $q_{\text{ж}} = 1 \text{ W/cm}^2$.

It is necessary to find the length of the condenser ~~capacitor of~~

$L_{\text{ж}}$:

$$L_{\text{ж}} = \left[\frac{\frac{\sigma}{R_{min}}}{\frac{\Pi^2}{\rho_{\text{ж}} \Pi^2} + \frac{\mu_{\text{ж}} \Pi}{2 \kappa_{\text{ж}}}} \right]^{1/2},$$

$$\Pi = \frac{q_{\text{ж}} r_i}{r' (r_i^2 - r_0^2)},$$

$$\Pi = \frac{10^4 \cdot 2 \cdot 10^{-2}}{1,1 \cdot 10^6 (0,39) \cdot 10^{-4}} = \frac{2}{1,1 \cdot 0,39} = 4,6 \frac{\text{Kz}}{\text{M}^3 \text{сек}},$$

$$L_{\text{ж}} = \left[\frac{18,3 \cdot 10^{-3}}{3 \cdot 10^{-4} \left(\frac{4,62}{0,79 \cdot 10^3 \cdot 0,49} + \frac{1,2 \cdot 10^{-2} \cdot 4,6}{6 \cdot 10^{-9} \cdot 2,07 \cdot 10^3} \right)} \right]^{1/2} =$$

$$= 31 \text{ cm.}$$

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Two-dimensional model

let us examine the model of ^athe flat ~~plate~~ porous core of the condenser ~~capacitor~~ of ^athe thermal ^{tube} ~~unit~~ or steam chamber (Fig. 6).

Let us assume: the thickness of core is small in comparison with the length of condenser ~~capacitor~~ (body of ~~the~~ semi-bounded ~~one~~ dimensions); condenser ~~capacitor~~ is separated from evaporator ~~capacitor~~ by the adiabatic zone whose length is considerably greater than the length of condenser ~~capacitor~~; the local heat flux, ~~which is~~ removed from condenser ~~capacitor~~, ^{is} ~~alternating~~ ^{and} variable depends on coordinate x ; the condensation of vapor occurs on the surface of porous core; the local condensation rate is determined by the rate of capillary absorption into porous core; the fluid flow in capillary-porous body ^{is} ~~laminar~~ ^{and} is determined by the law of Darcy, the rate of flow of liquid is accepted ~~the~~ ^{to} equal average rate of flow of liquid in pores; the effect of gravitational field we disregard.

The law of conservation of mass in porous core under stationary conditions takes the form

$$\frac{d^2 p}{dx^2} = 0. \quad (1.138)$$

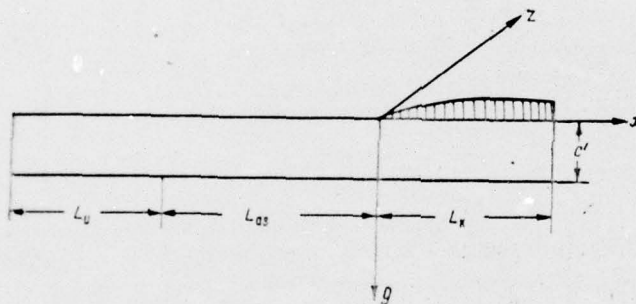


Fig. 6. Diagram of the core of thermal tube.

Boundary conditions:

1) on the internal surface of porous core

$$\begin{aligned} y=0, P_m &= P_n \text{ for } x \geq 0, \\ \frac{dP_m}{dy} &= 0 \text{ for } L_a < x < 0, \\ P_m &= P_n \text{ for } x \leq -L_a; \end{aligned} \quad (1.139)$$

2) on the external surface of $y = C$, $dp/dy = 0$ for all x ; P_n and P_n - the pressure of liquid on the surface of core in evaporator ~~capacitor~~ and condenser ~~capacitor~~ of thermal ~~capacitor~~ tube.

Pressure ~~capacitor~~ ^{of vapor} above the surface of the condenser ~~capacitor~~ ^{of vapor} P_n is constant. Pressure ~~capacitor~~ ^{of vapor} in evaporator ~~capacitor~~ can turn out to be variable, especially if the process of evaporation is realized from the zone of sinking.

In this analysis this we disregard, since the condenser ~~evaporator~~ and evaporator ~~evaporizer~~ are divided by the adiabatic zone of a sufficient extent. We will consider that pressure ^{of vapor} ~~in~~ in evaporator ~~evaporizer~~ ^{is} constant ^{to} and equal ^{to} the average value of the pressure of P_v^* .

the solution to equation (1.138) with boundary conditions (1.139) takes form [63]

$$\begin{aligned}
 U_m(x) &= U_m(x, 0) = \\
 &= \frac{\pi}{4} \cdot \frac{K_1}{C} \cdot \frac{P_v - P_v^*}{P_m} \cdot \frac{1}{M(\alpha)} \times \\
 &\times \frac{\sqrt{2} \cos(a/2)}{[\cosh(X+a) - \cosh a]^{1/2}}, \quad (1.140) \\
 X &= \frac{\pi x}{C}, \quad a = \frac{\pi L_a}{2C}, \quad \alpha = \operatorname{tgh}(a/2),
 \end{aligned}$$

where $M(\alpha)$ - first-order complete elliptic integral with ~~modulus~~ modulus α . As can be seen from equation (1.140), rate change near $x = 0$ occurs according to the law of $x^{-1/2}$.

General ~~fluid~~ fluid flow through the porous core

$$\begin{aligned}
 j_m &= \int_0^\infty \rho_m U_m(x) b dx = \frac{\rho_m b K_1 (P_v - P_v^*)}{2\mu_m} \times \\
 &\times \frac{M\sqrt{1-\alpha^2}}{M(\alpha)}. \quad (1.141)
 \end{aligned}$$

If $a > 5$, then a good approximation of equation (1.140) ~~is~~ will be

$$U(x) = \frac{\pi}{4} \cdot \frac{K_1}{C} \cdot \frac{(P_n - P_n^*)}{\mu_n} \cdot \frac{1}{M(\alpha)} \times \\ \times (\exp X - 1)^{-1/2}. \quad (1.142)$$

As can be seen from (1.142), with $x > c$, the rate of the motion of the liquid ~~is~~ $U_n(x)$ decreases according to the law $\exp(-X/2)$.

If one assumes that the amount of heat, transferred along ~~the~~ ^{tube,} is equal to the product of fluid flow ~~by~~ ^{by} heat of vaporization, then

$$Q = j_n r' = \frac{\pi}{4} \cdot \frac{\rho_n r' b K_1 (P_n - P_n^*)}{\mu_n M(\alpha)} \quad (1.143)$$

with $L_a \gg C$, when $\alpha \approx 1$.

From equation (1.143) the velocity ~~is~~ $U_n(x)$ can be defined as

$$U_n(x) = \frac{Q}{\rho_n r' C b} (\exp X - 1)^{-1/2}. \quad (1.144)$$

Heat exchange in the porous core of condenser ~~is~~ The equation of thermal conductivity takes the form

$$\frac{d^2T}{dy^2} = 0. \quad (1.145)$$

The boundary conditions:

with $y = 0$

$$-\lambda_\phi \frac{\partial T}{\partial y} = \rho_m U_m(x) r' \quad \text{for } x > 0,$$

$$\frac{\partial T}{\partial y} = 0 \quad \text{for } x < 0,$$

with $y = C$

$$T = T_\kappa \quad \text{for } x > 0,$$

$$\frac{\partial T}{\partial y} = 0 \quad \text{for } x < 0.$$

(1.146)

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The solution to equation (1.145) with boundary conditions (1.146) we will search for in dimensionless form

$$\Theta(X, Y) = (b\lambda_\phi/Q) [T_{(x,y)} - T_K], \quad (1.147)$$

where X and Y - dimensionless coordinates,

$$X = \frac{\pi x}{C}, \quad Y = \frac{\pi y}{C},$$
$$\frac{\partial \Theta}{\partial Y} = f(x) \equiv -\frac{1}{\pi} (\exp x - 1)^{-1/2}. \quad (1.148)$$

The mixed boundary conditions. Let us make a convolution by the conformal conversion of the unbounded medium into that which was semi-bounded with the aid of substitution

$$\sin \xi = \exp z,$$

where $z = X + iY$, $\xi = \xi + i\eta$.

thus, boundary-value problem for $\Theta(\xi, \eta) = \Theta(X, Y)$ is reduced to

$$\frac{\partial^2 \Theta}{\partial \xi^2} + \frac{\partial^2 \Theta}{\partial \eta^2} = 0, \quad (1.149)$$

$$\xi = -\frac{\pi}{2}, \quad \Theta = 0 \quad \text{for} \quad \eta > 0,$$

$$\eta = 0, \quad \frac{\partial \Theta}{\partial \eta} = 0 \quad \text{for} \quad -\frac{\pi}{2} \leq \xi \leq \frac{\pi}{2},$$

$$\xi = \frac{\pi}{2}, \quad \frac{\partial \Theta}{\partial \xi} = F(\eta) = \frac{\sqrt{2} \cosh(\eta/2)}{\pi \cosh \eta} \quad \text{for} \quad \eta > 0.$$

The solution we obtain by cosine Fourier transform. If

$$\bar{\Theta}(\xi, \lambda) \equiv \sqrt{\frac{2}{\pi}} \int_0^{\infty} \Theta(\xi, \eta) \cos \lambda \eta d\eta,$$

then equation (1.149) ~~stops~~ ^{becomes}

$$\frac{\partial^2 \Theta}{\partial \xi^2} - \lambda^2 \bar{\Theta} = 0 \quad (1.150)$$

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with boundary conditions

$$\begin{aligned} \xi = -\frac{\pi}{2}, \quad \bar{\Theta} &= 0, \\ \xi = \frac{\pi}{2}, \quad \frac{\partial \Theta}{\partial \xi} &= \sqrt{\frac{2}{\pi}} \cdot \frac{\cosh \left(\lambda \frac{\pi}{2} \right)}{\cosh \pi}. \end{aligned} \quad (1.151)$$

The solution for $\bar{\theta}$

$$\bar{\theta}(\xi, \lambda) = \sqrt{\frac{2}{\pi}} \frac{\sin h\lambda \left(\xi + \frac{\pi}{2} \right) \cos h \left(\lambda \frac{\pi}{2} \right)}{\lambda \cos h^2 \lambda \pi} \quad (1.152)$$

and respectively, if we pass back ~~back~~ to θ ,

$$\theta(\xi, \eta) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{\infty} \bar{\theta}(\xi, \lambda) \exp(-i\lambda\eta) d\lambda. \quad (1.153)$$

In the final form

$$\begin{aligned} \Theta(\xi, \eta) = & \frac{1}{\pi^2} \sum_{m=0}^{\infty} \left\{ \left[1 + \left(m + \frac{1}{2} \right) \eta \right] \times \right. \\ & \times \left[(-1)^m \cos \left(m + \frac{1}{2} \right) \xi + \sin \left(m + \frac{1}{2} \right) \xi \right] + \\ & + \left(m + \frac{1}{2} \right) \left[(-1)^m (\pi + \xi) \sin \left(m + \frac{1}{2} \right) \xi - \right. \\ & \left. \left. - \xi \cos \left(m + \frac{1}{2} \right) \xi \right] \right\} \frac{\exp \left[- \left(m + \frac{1}{2} \right) \eta \right]}{\left(m + \frac{1}{2} \right)^2}, \quad (1.154) \end{aligned}$$

which is correct as the solution to equation (1.149) for $\eta > 0$.

In order to pass to x and y , $\frac{x}{\lambda}$ is convenient to use formulas

$$\exp 2x = \sin^2 \xi + \sin^2 h^2 \eta,$$

$$\operatorname{tg} Y = \operatorname{ctg} \xi \operatorname{tg} h \eta.$$

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Figure 7a shows isotherms and adiabatic curves in porous core in the zone of condensation.

For $Y = \pi/2$, $\xi = 0$ equation (1.154) is reduced to form

$$\begin{aligned} \Theta(0, \eta) = \frac{1}{\pi^2} \sum_{m=0}^{\infty} \frac{(-1)^m}{\left(m + \frac{1}{2}\right)^2} \left[1 + \left(m + \frac{1}{2}\right) \eta \right] \times \\ \times \exp \left[- \left(m + \frac{1}{2}\right) \eta \right], \quad (1.155) \end{aligned}$$

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HEAT-TRANSMITTING TUBES, (U)

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where ~~the~~ $\eta = \sinh^{-1} \exp x$.

series (1.155) ~~they~~ converge for all η , including $\eta = 0$, ($x = -\infty$), where is obtained value 3.664.

The temperature in the adiabatic zone of core (Fig. 7b) is equal to

$$T(-\infty, y) = T_{\kappa} + 0,371 \frac{Q}{b\lambda_{\Phi}}. \quad (1.156)$$

Temperature in point (0, 0)

$$T(0, 0) = T_{\kappa} + 0,890 \frac{Q}{b\lambda_{\Phi}}. \quad (1.157)$$

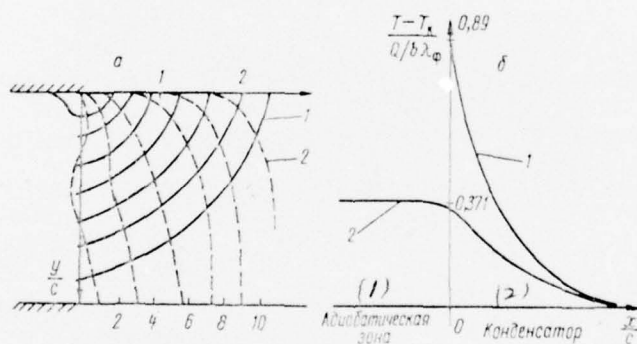


Fig. 7. Isotherms and adiabatic curves in porous core in the zone of condensation (a): 1 - isotherm; 2 - adiabatic curve; the temperature in the adiabatic zone of core (b): 1 - the temperature of the surface of core; 2 - the temperature in the middle zone of core.

Key: (1). Adiabatic zone. (2). Condenser.

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4. Calculation of the process of heat transfer in the thermal ^{tube} ~~duct~~ or ~~the~~ steam chamber in the presence of the boundary conditions of the 1st kind ^{on} ~~to~~ external surface.

Let on the external surface of the evaporator/~~vaporizer~~ of the thermal ^{tubes} ~~duct~~ of L_n be ^{maintained} ~~supported~~ the temperature of T_n , and on the external surface of the condenser/~~capacitor~~ of L_k - ~~the~~ temperature of T_k . It is necessary to determine the heat flux, transferred along ^{tube} ~~duct~~.

Figure 6 shows ^a ~~the~~ diagram of the core of thermal ^{tube} ~~duct~~. Let us

assume that ~~the~~ $L_a \gg L_m > L_n$ and that heat is transferred through the porous core to ^{the} wall by means of thermal conductivity; then

$$\frac{Q}{L_n b} = \lambda_{\phi}^n \frac{T_n - T_{nac}}{C}, \quad (1.158)$$

$$\frac{Q}{L_n b} = \lambda_{\phi}^k \frac{T_{nac} - T_k}{C}, \quad (1.159)$$

$$T_{nac} = \frac{L_n \lambda_{\phi}^n T_n + L_n \lambda_{\phi}^k T_k}{L_n \lambda_{\phi}^n + L_n \lambda_{\phi}^k}. \quad (1.160)$$

If we use the results ^{of} [63], then fluid flow ^{along} the porous core of thermal ~~duct~~ ^{tube} can be found in the form

$$j_m = \int_0^{\infty} \rho_m U_m(x) b dx = \frac{\rho_m b K (\Delta P_m)}{2 \mu_m} \cdot \frac{M \sqrt{1 - \alpha^2}}{M(\alpha)}, \quad (1.161)$$

$$U_m(x) = \frac{\pi}{4} \cdot \frac{K}{C} \cdot \frac{\Delta P_m}{\mu_m} \cdot \frac{1}{M(\alpha)} \cdot \frac{\sqrt{2} \operatorname{ch}(a/2)}{[\operatorname{ch}(X+a) - \operatorname{ch} a]^{1/2}}, \quad (1.162)$$

$$\Delta P_m = P_{mk} - P_{mn},$$

$$X = \frac{\pi x}{C}, \quad a = \frac{\pi L_a}{2C}, \quad \alpha = \operatorname{tg} h(a/2).$$

For $a > 5$

$$U_m(x) = \frac{\pi}{4} \cdot \frac{M}{C} \cdot \frac{\Delta P_m}{\mu_m} \cdot \frac{1}{M(\alpha)} \cdot \frac{1}{\sqrt{\exp x - 1}}, \quad (1.163)$$

where $\alpha \approx 1$; $M(\alpha)$ is first-order complete elliptic integral with ~~module~~/modulus α .

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The maximum amount of heat, transferred along thermal ~~duct~~^{tube} during the evaporation of liquid from the surface of porous core, can be found from formula

$$Q_{\max} = j_{\max} r', \quad (1.164)$$

j_{\max} ~~is~~ is determined from formula (1.161), when $\Delta P_{\text{ac}} = \frac{2\sigma}{R_{\text{min}}}$.

The unknown amount of heat, transferred along thermal ~~duct~~^{tube} in the presence of ~~the~~ constant temperature ~~of~~ T_u on the external surface of evaporator/~~vaporizer~~ and T_k on the external surface of condenser/~~capacitor~~, can be obtained, according to molecular-kinetic theory, in the form

$$Q = jr' = r' A \Pi \sqrt{\frac{m}{2\pi K T_{\text{nac}}}} P^*(T_{\text{nac}}). \quad (1.165)$$

But in this formula the coefficient of evaporation A to us is unknown, ^{and} it can change over wide limits depending on the properties

of liquid.

If one assumes that the coefficient of evaporation A ^{does} not depend very greatly on temperature, ~~but~~ ^{and} the temperature of T_{nac} ^{is} ~~it is~~ little affected depending on T_H and T_K , then it is possible to write the following equality:

$$Q_{max} = j_{max} r' = r' A \Pi \sqrt{\frac{m}{2\pi k T_{nac}^{max}}} p^*(T_{nac}^{max}), \quad (1.166)$$

then the ^{ratio} ~~relation~~ of Q/Q_{max} will not contain the coefficient of evaporation A

$$\frac{Q}{Q_{max}} = \sqrt{\frac{T_{nac}^{max}}{T_{nac}}} \cdot \frac{p^*(T_{nac})}{p^*(T_{nac}^{max})}. \quad (1.167)$$

Knowing the values of T_H and T_K , we find T_{nac} . If we calculate Q_{max} according to formula (1.166), then it is possible taking into account (1.167) to find Q ~~by~~ knowing the value of T_{nac} and T_{nac}^{max} .

The value of $T_{\text{nac}}^{\text{max}}$ is determined with the aid of the ~~curve~~/graph of the dependence of Q_{max} on T_{κ} ^{at} ~~with~~ different T_{H} or Q_{max} on T_{κ} ^{at} ~~with~~ different T_{κ} in formula

$$Q = Q_{\text{max}} \left(\frac{T_{\text{nac}}^{\text{max}}}{T_{\text{nac}}} \right)^{1/2} \cdot \frac{P^*(T_{\text{nac}})}{P^*(T_{\text{nac}}^{\text{max}})} \quad (1.168)$$

This analysis is valid when Q is always less than ~~the~~ Q_{max} .

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Chapter 2.

STUDY OF PROCESSES HEAT- AND MASS EXCHANGE IN THERMAL ^{TUBES} ~~DUCTS~~.

1. Determination of capillary pressure head, permeability and porosity of capillary-porous bodies during the motion of liquid along pores.

For the study of processes heat- and ~~a~~ mass exchange in capillary-porous heat exchangers ⁱⁿ ~~with~~ the presence of phase transitions, arose the need for selecting the appropriate class of materials and conducting detailed investigation along with thermophysical ~~the~~ hydrodynamic and structural characteristics of specimen/samples.

^{By} Under the transport properties of capillary-porous bodies, is implied permeability K and the capillary pressure head ~~of the~~ ΔP_c , which can be also expressed by the maximum ^{height} ~~altitude~~ of the capillary elevation of the liquid ~~of~~ h_{max} . Knowledge of the transport properties of capillary-porous bodies is necessary for the calculation of the parameters of the ^{operation} ~~work~~ of thermal ^{tubes} ~~ducts~~, evaporative porous heat exchangers, condenser/capacitors, capillary pumps, etc. Since ~~the~~ real capillary-porous bodies ^{have pores} ~~porous~~ and ~~the~~ capillaries of different ^{sizes} ~~size/dimensions~~, usually to evaluate ^(Properties of a porous body are used the integral values) transport ~~as ranks~~ K and ~~an~~ ΔP_c , ^(which are) characteristic for the investigated material and determined experimentally.

In the composition of porous materials, ^{cermets obtained} ~~were the ceramic metal~~, ^{by sintering} ~~sintered of~~ powder, shavings, ^{tangled} ~~jumbled~~ wire, and also ~~the~~ packages of wire gauze. As metals were utilized nickel, stainless steel, bronze, titanium, copper, ^{and} brass.

Besides metallic porous materials, ~~by us~~^{we} was investigated a series of electrical insulators. They include glass cloth ASTT- 6-5-2, E-0,1, sintered fiberglass ~~of 30-1~~^{2hS-1} with the parallel and universal laying of filament, sintered powder of Etakril, quartz sand, fireclay ceramics, ^{and} powder Al_2O_3 .

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The determination of capillary pressure head and permeability of porous bodies was ~~realize~~/accomplished during the motion along them of a series of the liquids: water, Freon, acetone, alcohol, gasoline, ^{and} liquid nitrogen.

For the investigation of the characteristics of porous materials pointed out above by us were utilized both the known, described in ~~the~~ literature methods and the method ^p developed in the laboratory of low temperatures [20] of ITMO the A.S. of the B.S.S.R.

From the investigated by us ~~ceramic-metal~~^{cermets} the best wettability possessed ~~the~~ porous nickel, especially oxidized.

Somewhat ~~more~~^{less wettable} ~~badly~~ were ~~wet the~~ stainless steel, titanium, copper.

Table 1.

(1) Материал	(1a) Пористость, 11%
(2) Спеченные никелевые сетки:	
(2a) 20 проволок на 1 см., диаметр проволоки 0,2 мм	62,5
(2a) 40 проволок на 1 см., диаметр проволоки 0,1 мм	67,9
(3) Спеченная керамика из никелевой стружки	
(4) Диаметр стружки:	
0,015 мм	86,8
0,018 мм	82,8
(5) Спеченная керамика из стальной стружки	
(4) Диаметр стружки:	
0,02 мм	91,6
0,04 мм	82,2
(6) Спеченная керамика из порошка никеля	
(6a) Диаметр частиц:	
150-300 мм	59,7
300-850 мм	47,7
(7) Спеченное стекловолокно ЖС-1	30
(8) Стеклоткань АСТТ-6-2	53
(9) Порошок никеля	
(6a) Диаметр частиц:	
0,1 мм	24,8
0,2 мм	22,8
0,3 мм	20,3
(10) Порошок этикрила	
(6a) Диаметр частиц 0,05 мм	40
(11) Кварцевый песок	
(6a) Диаметр частиц 0,2 мм	36
(12) Порошок	
(6a) Диаметр частиц 0,01 мм	70

Key: (1). Material. (1A). Porosity, Π o/o. (2). Sintered nickel grids. (2a). ~~the~~ --- of thin wires to 1 cm., wire size of --- mm. (3). Sintered ceramics from nickel shaving. (4). Diameter of shaving. (5). Sintered ceramics from steel shaving. (6). Sintered ceramics from the powder of nickel. (6a). Diameter of particles. (7). Sintered fiberglass ^{2h} JS-1. (8). Glass cloth ASTT-b-2. (9). Powder of nickel. (10). Powder of Etakril. (11). Quartz sand. (12). Powder.

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Table 1 shows the porosity of the used in experiments cermet materials and electrical insulators. The determination of the ^{mean} ~~middle~~ porosity and the distribution of pores according to ~~a~~ radius of cermet materials ^{as} ~~were~~ conducted by the method of mercury porosity measurement ^{according} to the procedure, described in [68]. One should ^{point out} ~~indicate the fact~~ that by us were investigated ~~the~~ porous materials, possessing ^{mainly} ~~in essence the~~ apparent porosity, equal to the ratio of the volume of the ~~being~~ ^{ing} communicated with each other pores to the total volume of the body:

$$\Pi = \frac{V - P/\rho}{V},$$

where V, P, ρ - volume, ~~the~~ weight and the density of porous body.

Distribution of pores ^{over} ~~according to~~ ~~a~~ radius. The presence in the core of ~~the~~ pores of different geometric dimensions is considered ^{by} ~~the~~ distribution function of pores according to size/dimensions. If we use the concept of the hydraulic given diameter of pores, i.e., if we assume that the pores take the form of the spheres whose volume is

equal ^{to} actual, and these spheres are characterized by ~~conditional~~ ^{arbitrary} diameters, it is possible to introduce distribution function $\alpha(r)$, determined by equation

$$dw = -\alpha(r) dr, \quad (2.1)$$

where dw is ~~the~~ ^{the} volume of the pores, which have radii from r to $r + dr$; w - the volume of all pores, which have radius r or ^{of} ~~is~~ ^{is} greater ^{than} r . The most adequate method for determining the distribution function of pores in ceramic metal is the method of mercury porosity measurement. ^{By} measuring the pressure, necessary ~~in order~~ to force mercury inside porous body, and determining the volume of mercury in pores, we find ^{of} curve the distributions of pores according to ~~a~~ radius.

When ^a ~~the~~ nonwetting liquid, such, as mercury, is ^{forced} ~~indented~~ into pores, it forms ~~the~~ menisci whose curvature is determined by ^{and shape} ~~size~~ dimensions, ~~the form~~ of pores and by the properties of material.

For the determined ~~size/dimension~~ of pores, the accompanying external pressure is determined by the equation of Laplace ^{ce} - Young

$$\Delta P = \sigma \left(\frac{1}{R_1} + \frac{1}{R_2} \right), \quad (2.2)$$

where ΔP - ~~a~~ pressure drop across interface ^{of} liquid - vapor; σ is ~~a~~ ^{the} coefficient of surface tension; R_1 and R_2 - the radii of curvature ^{of} necessary for the description of three-dimensional surface in space.

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If we preliminarily pump out gas from ^{pores of} ~~about~~ core, and then ^{force in} ~~to indent~~ liquid, ~~then~~ $P \approx \Delta P$ when the partial vapor pressure of liquid is small.

Undoubtedly, real ^{shape} ~~form~~ and ~~the size/dimensions~~ of pores are ^{far} ~~different~~ from spherical or cylindrical; however, frequently ^{is} ~~they~~ used the equivalent radius of pores, which is defined as

$$r = 2 \frac{S}{L}, \quad (2.3)$$

where S is ~~a~~ cross-sectional area of pore; L - the perimeter of pore

$$F_{\sigma} L = PS, \quad (2.4)$$

F_{σ} - the force of surface tension per ~~the~~ unit of ~~the~~ length of interface:

$$F_{\sigma} = \sigma \cos \Phi, \quad (2.5)$$

Φ - the angle of contact ^{of} liquid - solid.

Combining
~~uniting~~ the given equations, we obtain

$$Pr = 2\sigma \cos \Phi. \quad (2.6)$$

For constant values of surface tension and angle of wetting, this equation directly connects ~~a~~ radius of pores r with pressure P , necessary for the extrusion of the nonwetting liquid into the porous medium.

In differential form equation (2.6) can be presented as

$$rdP + Pdr = 0. \quad (2.7)$$

Combination of equations (2.1), (2.6) and (2.7) gives expression for determining the distribution function of pores according to radius

$$\alpha(r) = \frac{P^2}{2\sigma \cos \Phi} \cdot \frac{dV}{dP}. \quad (2.8)$$

For determining $\alpha(r)$ it is necessary to measure P and V . After calculating $\alpha(r)$, according to formula (2.8) we can find r , knowing σ and the angle of Φ . For mercury $\sigma = 473$ dyn/cm at 20°C, ^{while} and the angle of $\Phi \approx 130^\circ$ (for metals). The derivative $\frac{dV}{dP}$ can be found from the curve/graph of dependence of P on V .

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From equation (2.6) is located the diameter of pores

$$D = \frac{4\sigma \cos \Phi}{P}. \quad (2.9)$$

Frequently instead of the hydraulic diameter of pores, they use the diameter of pores in the form of the median ~~of~~ D_M . It is defined as the ^{typical} ~~significant~~ dimension of pore, when 50c/o of pore ~~it~~ is filled by liquid.

Ratio of the flow area of the pores of porous core to ~~common/general~~/total cross section. The ratio of flow area to the overall cross-sectional area of porous body is defined as

$$F = \frac{S_{\text{nop}}}{S_{\text{obm}}} \quad (2.10)$$

In this case, it is assumed ~~to be~~ that the body is isotropic and that the size ~~dimensions~~ of pores, ^{through} ~~over~~ which moves the liquid, is substantially less than the size ~~dimensions~~ of body.

Let us examine porous body in the form of beam with a length of x_1 in direction x ^{and} ~~by~~ the cross-sectional area ~~of~~ S_{06m} in direction, perpendicular ^{to} x .

From equation (2.10) the area of pores is ^{sound} ~~located~~ as

$$S_{nop} = FS_{06m}, \quad (2.11)$$

$$\Pi = \frac{V_{nop}}{V_{\text{теша}}}.$$

The total volume of body can be ^(e)presented in the form

$$V_{\text{теша}} = \int_0^{x_1} S_{\text{теша}} dx. \quad (2.12)$$

~~It is~~ analogous/y,

$$V_{nop} = \int_0^{x_1} FS_{06m} dx. \quad (2.13)$$

Then porosity

$$\Pi = \frac{\int_0^{x_1} F S_{\text{osm}} dx}{\int_0^{x_1} S_{\text{osm}} dx} \quad (2.14)$$

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If we consider that the pores are distributed evenly throughout porous body and their size~~dimensions~~ are negligibly small in comparison with the size~~dimensions~~ of body, then it is possible to assume that the ratio of the flow area of pores to the ~~common/general~~ total cross section F , i.e., surface porosity, will be equal ^{to} volumetric porosity

$$F = \Pi. \quad (2.15)$$

Nonstationary method of the complex determination of porosity, capillary pressure head and permeability of dielectric porous bodies. For the investigation of the hydrodynamic and structural properties of nonmetallic capillary-porous bodies by us was developed the ~~unsteady~~ ^{unstationary} method, based on recording the fields of the concentration of liquid and velocity of its absorption in porous body against gravitational forces. This method makes it possible to define the following parameters:

a) the volume of pores per unit of volume of the porous body $\omega(r)$ and the distribution of pores ~~according to~~ ^{over} radius dw/dr ;

b) the ~~common/general/total~~ porosity ~~of A_{total}~~ ^{Π} ;

c) the maximum ~~altitude~~ ^{height} of the capillary elevation ~~of h_{max}~~ ^{h_{max}} ;

d) effective permeability as function of the concentration of the liquid ~~of~~ $K(u)$;

e) the ~~common/general/total~~ permeability K of porous body.

~~By~~
~~Under~~ the kinetics of the absorption of liquid against gravitational forces, one should to understand the dependence of the field of concentration of liquid in porous specimen/~~sample~~ on time $U(x, t)$. During the experimental study of the kinetics of absorption, was utilized the electrical capacitance method of the measurement of the local concentrations of liquid [67] by height of specimen/~~sample~~ into process of its absorption. ~~Fundamental installation diagram~~ ^{Schematic (of assembly)} is represented in Fig. 8a. The concentration of liquid in ~~the~~ ^a porous core, manufactured from glass cloth and arranged ~~located~~ vertically, was measured with the aid of 18 ~~condenser~~ capacitors. The ~~overall~~ ^{common} plate ~~for~~ ^{on} all capacitors was the metallic cylinder ~~to~~ which was coiled the glass cloth. The wire rings, which fix fabric, simultaneously ~~served as~~ ^{served as} ~~were~~ the second plates of capacitors. This form of porous specimen/~~sample~~ is analogous to the form of the cores of thermal tubes; therefore procedure makes it possible to determine directly the properties of these cores, without destroying them.

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The kinetics of the absorption of water by capillary-porous core from glass cloth ASTT-b-2 (7 layers, 3 mm), measured by this ~~installation~~ ^{assembly}, is given in Fig. 8b and 10.

From the curves, corresponding to ~~airfoil~~ ^{of} profile [^] concentration

of liquid, ~~on~~ ^{after} completion of the process of the absorption of $U(h)_{t \rightarrow \infty}$ (Fig. 8b), after converting according to formula

$$w = \frac{\gamma_0}{\gamma_M} U, \quad (2.16)$$

we obtain the integral and differential distributions of pores in capillary-porous core (Fig. 9). From them it is possible to obtain ~~the~~ information about a minimum radius of the pores ~~of~~ $r_{\min} = 28.4 \mu\text{m}$ and the ~~average~~/mean or predominant radius ~~of~~ $r_{\text{срел}} = 30 \mu\text{m}$ and ~~of the~~ porosity ~~of~~ $\Pi = 53\%$.

The maximum ^{height} ~~altitude~~ of capillary absorption, determined experimentally from the ~~airfall~~/profile of the concentration of liquid, was equal to 52 cm.

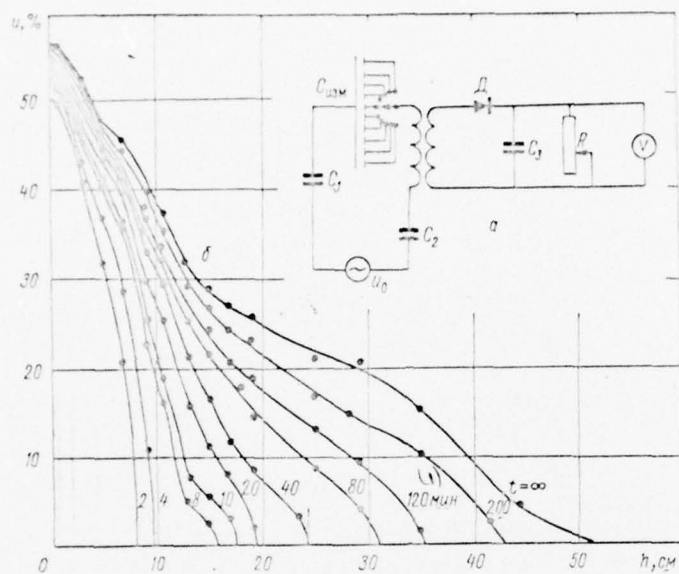


Fig. 8. Diagram of experimental installation according to the study of the motion of liquid in core against the forces of gravitation by electrical capacitance method (a); the kinetics of absorption (b).

Key: (1). min.

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Figure 10 depicts the kinetics of the front of the absorption of water in porous specimen/~~sample~~ and the dependence of the velocity of the motion of front ~~from~~^{on} height/~~altitude~~. It should be noted that the graphic dependence $U = f(1/h)$ it is not possible to approximate by the straight line which usually is utilized for determining the maximum ~~altitude~~^{height} of absorption. This is the consequence of the fact that ~~the~~ porous specimen/~~samples~~ made of glass cloth ~~are~~ have semi-capillary structure.

The kinetics of the absorption of liquid in porous body against the forces of gravitation makes it possible to determine under unsteady conditions the effective or dynamic permeability ~~of the~~ K_{dyn} which characterizes the permeability of that part of the capillaries which absorbs liquid at ~~the~~^a given instant at base ~~altitude~~^{height}. Since in this case the discussion concerns absorption, ~~under~~^{by} concept dynamic penetrability one should understand permeability only ~~to~~^{of} that ~~the~~ stops of capillaries, ~~that~~^{which} at this torque/moment participates in the formation/~~education~~ of the front of impregnation.

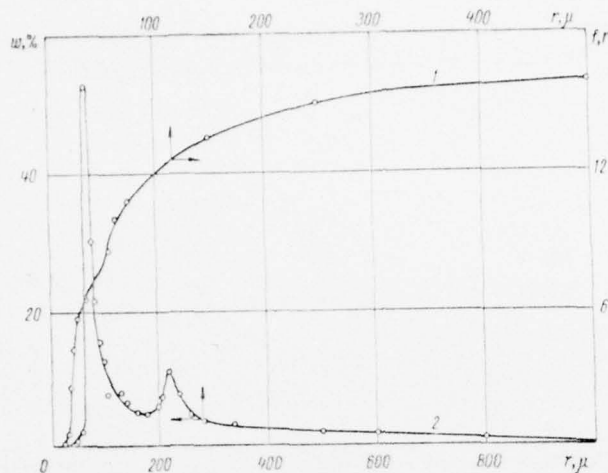


Fig. 9. Integral (1) and differential (2) distribution of pores in the capillary-porous core of thermal tube.

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On the other hand, $K_{\text{дин}}$ ~~there~~ ^{is a} certain modification of the coefficient of ~~the~~ capillary conductivity of ~~the~~ α_{ψ} , which enters the flow equation during free absorption [68]

$$i = \alpha_{\psi} \nabla \psi. \quad (2.17)$$

~~By~~ Utilizing ~~the~~ ^{the} law of Darcy, it is possible to obtain the flow equation during free absorption in the form

$$v = K_{\text{дин}} \frac{\gamma_{\text{ж}}}{\gamma_0} \cdot \frac{\frac{2\sigma \cos \Theta}{r_{\text{min}}} - h}{\mu h U_{\text{дин}}},$$

where the coefficient ~~of~~ K_{dum} it is ^a ~~the~~ function of height/altitude h .

The application ~~use~~ of ~~a~~ law of Darcy ^{to} ~~for~~ our case is justified by the fact that in ~~the~~ literature ^{there} is information about the applicability of this law for unsteady processes [70]. Since with an increase in ^{height} ~~altitude~~ of specimen/~~sample~~ decreases ~~a~~ radius of ~~the~~ pulling capillaries and their ~~portion~~/fraction also ~~they will be~~ changed, according to the distribution of pores ^{over} ~~according to~~ radii, the dependence ~~of~~ $K_{\text{dum}}(h)$ can have several maximums.

Figure 11 gives ^a ~~the~~ section of this curve for our porous specimen/~~sample~~.

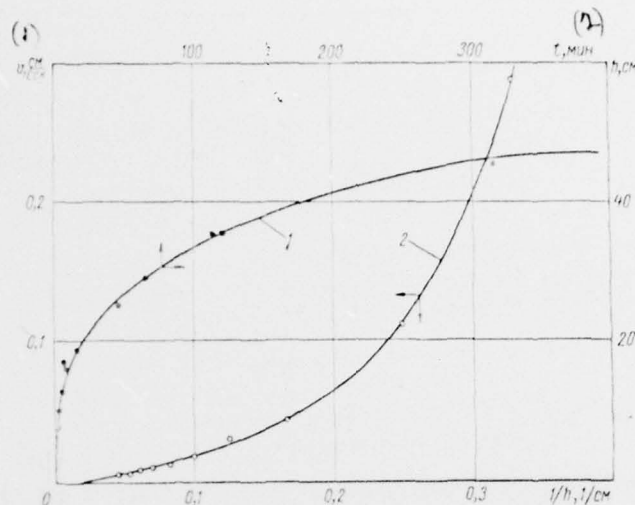


Fig. 10. The kinetics of the front of impregnation in core $u = f(h)$ (is curve 1) and the dependence of the velocity of the motion of front ~~from~~ ^{on} height/altitude (is curve 2).

Key: (1). cm/s. (2). t, min.

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The integral permeability of specimen/sample can be obtained by the graphical integration of the dependence of K_{diff} for formula

$$K(r) = \frac{1}{r - r_{\text{min}}} \int_{r_{\text{min}}}^r K_{\text{diff}}(r) dr. \quad (2.18)$$

This integral permeability $K(r)$, whose section is given in Fig. 11, must have a form, analogous of the integral distribution curve of pores ^{over} according to radii, i.e., for ~~at~~ $r \rightarrow r_{\max}$, $K(r) \rightarrow K$, where K - is the total permeability of specimen/sample.

In formula (2.18) enters radius r , which is unambiguously connected ^{with} ~~at~~ the velocity of the motion of the front of absorption, i.e., this ^(is the) radius of that ~~the~~ stops of the capillaries which form the front of absorption. Graphically, the dependence of this radius of capillaries on the height/altitude of front is given in Fig. 12a (curve 2), for calculation of which were utilized the curves of the kinetics of absorption in coordinates $U(h)/t$ (see Fig. 8b). On each ~~airfoil~~/profile of the concentration of liquid, which corresponds to the defined point in time, amount of liquid at the level of the front of impregnation can be approximated as certain finite quantity of concentration.

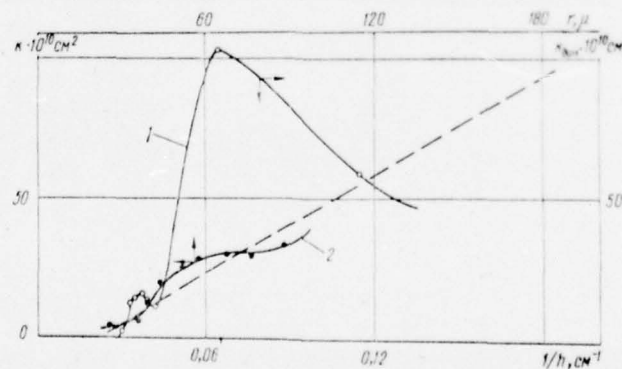


Fig. 11. Dependence of dynamic permeability (1) ~~of~~ $K_{dyn} = f(h)$ and the dependence of integral permeability (2) $K = f(r)$.

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By utilizing curve of the integral distribution of pores and the final instantaneous values of the concentration of the liquid of u_{MCH} ~~aaaaaa~~ enumerated above, it is possible comparatively simply to determine ^{the} radius of the capillaries which contain liquid in the amount, which corresponds to this instantaneous value of concentration. This characteristic, presented in Fig. 12a ~~is~~ (curve 1), is interesting ^{more} ~~even thereby~~ in that it makes it possible to judge the value of the angle of wetting during absorption.

In ~~the~~ literature there is data [72] about the fact that ^{during} ~~with~~ ~~by~~ absorption the dynamic angle of wetting decreases, i.e., the cosine of angle increases. In [73], it is said that dependence $\cos \theta (h)$ is ^a ~~the~~ straight line, analogous to straight line, given ⁱⁿ ~~on~~ Fig. 12b ~~is~~ (curve 2).

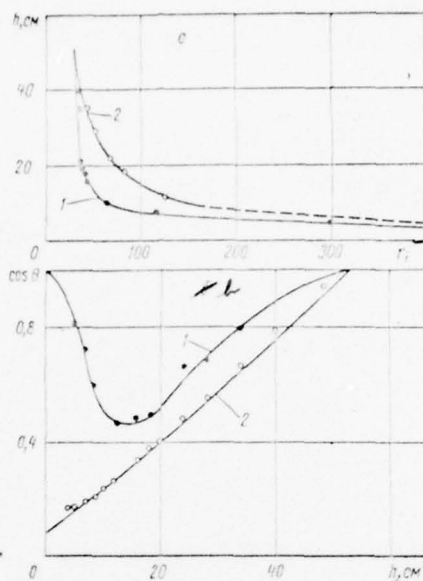


Fig. 12. Dependence of r radius of the capillaries, forming the front of impregnation, on height/altitude (a): 1 - experimental; 2 - theoretical; θ change of the dynamic angle of wetting in capillary porous core ^(depending on) from height/altitude (b): 1 - [20]; 2 - [73].

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In this work during the calculation, is utilized the simplified equation of ^{Navier} ~~Navier~~ - Stokes for capillaries in the form

$$\cos \theta = \frac{4\mu h}{r\sigma} \cdot \frac{dh}{dt} + \frac{rgh\gamma_{pc}}{2\sigma} \quad (2.19)$$

In this expression ~~r~~ ^{the} ~~it~~ is a minimum radius of the pores of r_{min} .
In actuality, radii ^{of} ~~the~~ stops of the capillaries, which participate in the formation/~~education~~ of the front of impregnation, are changed, which is shown Fig. 12b (~~is~~ curve 1) for $\cos \theta$ (h). This dependence can be explained by the fact that the porous body has pores of quite

different size/~~dimension~~. For the capillaries ^{with} of a large radius, the velocity of absorption is sufficiently great, and the height of absorption is small; therefore, at the moment of absorption ^(the formula for) meniscuses in the capillaries, forming the front of impregnation, will be ~~in~~ ~~formula~~ close to static.

For determining h_{\max} and ΔP_R , besides electrical capacitance method, by us widely were utilized the method of γ -radiation, the method of litmus paper slips, the method of electrical resistance and the method of ~~the break~~ ^{ing} of liquid column. The applicability of one method or the other ^{resulted from} ~~was explained by~~ the special ^{the} feature/~~peculiarities~~ of capillary-porous body and liquid. So, ^{the} ~~a~~ method of litmus paper slips and ^{the} ~~a~~ method of ~~the break~~ ^{ing} of liquid column ^{are} ~~is~~ conveniently utilized for determining h_{\max} and ΔP_R of metallic porous materials.

When using ~~a~~ method of litmus paper slips [6] and of ~~the breaking~~ of liquid column [31] for determining h_{\max} and ΔP_R , we utilized the known formulas, published in ~~the~~ literature. The effective motive power for raising ~~ion of~~ liquid in capillary is equal to

$$\Delta P_R = \frac{2\sigma \cos \theta}{R_R} . \quad (2.20)$$

Gravitational force, which blocks ^{raising} ~~the elevation~~ of liquid in capillary

$$\Delta P_g = \rho g x \sin \alpha. \quad (2.21)$$

Force of friction, which has effect during the motion of liquid along capillary

$$\Delta P_{rp} = \frac{\mu_m j_{mx}}{K_{\rho_m} A_S}. \quad (2.22)$$

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Under stationary conditions ^{at} ~~with~~ the maximum fluid flow for the capillaries ~~of~~ j_{max} , the pressure of liquid because of capillary forces is balanced by force of friction and by gravitation

$$\frac{2 \cos \theta \sigma}{R_H} = \rho_m g x \sin \alpha + \frac{\mu_m j_{mx}}{K_{\rho_m} A_S}, \quad (2.23)$$

where ~~the~~ A_s - cross-sectional area of the pores, filled by liquid.

In this expression ^{the} unknowns are the R_K and K . It is assumed that the angle of wetting θ we know. Usually it is close to 0. If it differs from zero, then is known ^{ratio} ~~relation~~ \cos of θ/R_K . As a rule, the properties of liquid are known, distance x is assigned, and the flow ~~of~~ j_m is determined experimentally. The maximum ^{height} ~~altitude~~ of the capillary ^{ascension} ~~elevation~~ of h_{max} we determine in ^{an} ~~the~~ atmosphere of saturated ^{steam} ~~pair~~ or in the atmosphere of air.

^{The} ~~At~~ effective radius of the pores of the porous core ~~of~~ R_K can be determined by formula (2.20)

$$R_K = \frac{2\sigma}{\rho_{\text{net}} g h_{\text{max}}} \quad (2.24)$$

The nominal radius of the pores of core is determined with the aid of microscope. Usually it is less than the effective diameter of pores.

It must be noted that when using grids as capillary cores, the capillary forces act equally both on the horizontal and on vertical lines.

In Table 2, obtained by us data are compared with data ^{from} ~~on~~ ΔP_R , published in ~~the~~ literature.

One should ^{point out an} ~~indicate the~~ interesting ~~special feature~~/peculiarity of the kinetics of the absorption of polar liquids in porous dielectrics during the ^{application} ~~imposition~~ of a DC field. We have recorded the considerable intensification of the process of the absorption of ethanol in porous Etakril during the ^{application an} ~~imposition~~ of electric field ^{with} ~~by~~ ^{of} intensity/strength 1 kV/cm [66].

Determination of permeability K of porous bodies.

Permeability can be determined by the forced ^{acceleration} ~~screw die~~ of the liquid through the porous body under the action of pressure gradient ^{and} by flow of liquid under the action of gravitational field.

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^{During} ~~With the forced~~ ^{acceleration} ~~screw die~~ of the liquid through the porous body, it is insulated from the environment by ^{placement} ~~location~~ into the airtight chamber. We have used both procedures for determining permeability K of a series of porous materials. In the first version the liquid is ^{accelerated} ~~driven off~~ under pressure through the specimen/~~sample~~ and is measured the pressure differential along the path of motion of liquid.

Such experiments are justified for the thick porous cores when the ~~effect of~~ wall effect can be disregarded. For ~~fine~~/thin cores, ^[it is inadvisable to determine] for example grids, permeability according to this procedure, ~~to determine is inexpedient~~ since the working conditions of core in tube are not equivalent to the conditions of experiment ^{for determining} ~~regarding~~ permeability.

During the determination of permeability one of the important parameters of porous body is the size/~~dimension~~ of pores. For example,

it is possible to assume that highly porous cores have larger permeability, than low porous ^(ity cores) with the identical size/~~dimension~~ of pores. However, if the size/~~dimension~~ of pores of low-porosity materials is ~~be~~ greater the size/~~dimension~~ of the pores of highly porous body, then it can ~~seem~~ ^{turn out} that the permeability of low-porosity material will prove to be ~~above~~ ^{higher} because of the ~~less~~ ^{lower} value of the coefficient of the crookedness of path.

^{The} method of the determination of permeability during flow of liquid through ~~the~~ ^a porous body under the action of ~~the~~ ^a field of gravitation we have used for ~~fine~~/thin porous cores, since this method makes it possible to leave the ~~exposed~~ surface of porous body ^(exposed) maintain a constant radius of curvature of interface ^{of} liquid - vapor (or gas). This method has an advantage - ^{it} makes it possible to define the permeability of porous body as ^a function of the radius of curvature of the interphase interface. When free interface ~~is~~ ^{is} of ~~present~~, liquid - ^{vapor} ~~pairs~~ constant radius of curvature along core it is possible to ensure by means of the ~~slope~~/inclination of core at the determined angle to horizontal, and also ^{by} regulating the density of fluid flow until ^{the} an incidence/drop in the fluid pressure ~~on the~~ ^{per} unit of length becomes equal to ^{the} a change in the pressure of liquid column ^{per} on ~~the~~ unit of length.

In this manner it is possible to investigate the effect of the depth of the meniscus of liquid on the coefficient of permeability.

Table 2.

1 Пористый фтиль	2 Жидкость	3 Окружающая атмосфера	4 Измеренное (4) максимальное ка- пиллярное давле- ние, кг/м ²	5 Расчетный эффек- тивный диаметр, мкм	6 Номинальный диаметр, мкм	7 Пористость, %	8 Толщина фтиля
(7) Сетка из нержавеющей стали (80 проволонок на 1 см)	(10) метанол	(4) пар	87	108	87	73,3	87
(12) То же	»	(13) воздух	87	111	87	73,3	
(14) Сетка из бронзы (80 проволонок на 1 см)	»	(13) воздух	87	105	7,2		
(15) Сетка из нержавеющей стали (80 проволонок на 1 см)	(10) бензин	(13) воздух	109	112	87	73,3	
(15) Сетка из нержавеющей стали (80 проволонок на 1 см)	(17) вода	(13) воздух	261	115	87	73,3	
(18) Сетка из бронзы (80 проволонок на 1 см)	»	(13) воздух	240	190	72		
(19) Сетка из никеля (80 проволонок на 1 см)	»	(13) воздух	240	123		67,6	
(20) Пористая медь (спеченная из порошка)	(21) метанол	(11) пар	21	240	200-1000	94,5	450
(21) То же	бензин (16)	(13) воздух	26,8	450	200-1000	94,5	450
»	(17) вода	(13) воздух	70	440	200-1000	94,5	450
»	»	(13) воздух	64	475	300-1000	91,2	2500
(22) Пористая медь (спеченная из стружки)	(21) метанол	(11) пар	16,7	550	200-800	89,5	2500
(22) То же	(16) бензин	(13) воздух	27,6	425	200-800	89,5	2500
»	(17) вода	(13) воздух	65	450	200-800	89,5	2500
(23) Пористый никель (спеченный из порошка)	(21) метанол	(11) пар	17,4	525	250-625	96	2500
(24) То же	(16) бензин	(13) воздух	27,6	425	250-625	96	2500
(14) То же	(17) вода	(13) воздух	65	450	250-625	96	2500
»	»	(13) воздух	65	450	450-1600	94,4	2600
(24) Пористый никель (спеченный из стружки)	(17) вода	(13) воздух	94	325	675		
(25) Пористая медь (спеченная из стружки)	дистиллирован- (26) ная вода	(13) воздух	101,6	44		80	9000
(27) Пористая медь окисленная (спеченная из стружки)	(28) то же	(13) воздух	362,9	40,6	46	80	9000
(29) Пористая медь (спеченная из стружки)	дистиллирован- (26) ная вода	(13) воздух	123,8	—	—	80	9000
(30) Пористая медь окисленная	(28) то же	(13) воздух	457	32,5	—	82	9000
(31) Пористая медь из спрессованного неспеченного порошка	»	(13) воздух	1568	9,4	—	52	9000
(32) Медная сетка (60×60 меш), диаметр проволоки 0,165 мм	»	(13) воздух	30,63	494	—	48	9000
(33) Никелевая сетка (50×50 меш), диаметр проволоки 0,102 мм	»	(13) воздух	8,13	1820	—	67	
(34) Никелевая сетка окисленная (50×50 меш), диаметр проволоки 0,102 мм	»	(13) воздух	25,4	580,2	—	67	
(35) Никелевая сетка (120×120 меш), диаметр проволоки 0,076 мм	»	(13) воздух	79,37	280	—	67	
(36) То же, окисленная	»	(13) воздух	79,37	190	—	67	
(37) Стеклоткань АСТТ-(6)-С-2 (10 слоев)	»	(13) воздух	280	106	—	61,8	4250
(38) Стеклоткань Э-0,1 (30 слоев)	»	(13) воздух	285	104,8	—	60	3000
(39) Стеклоткань никель, фр. 0,2	(42) ацетон	(13) воздух	56	212	—	33,5	3000
(40) Стеклоткань Э-0,1 (24 слоя)	дистиллирован- (26) ная вода	(13) воздух	350	84	—	64,5	2000
(41) Спеченное стекловолокно ЖС-1 с параллельной укладкой	(28) то же	(13) воздух	310	96	—	54,4	
(42) Латунная сетка (четырёхъячеечная, 23 слоя)	(13) ацетон	(13) воздух	260			72,4	6000

Key: (1). Porous core. (2). Liquid. (3). Surrounding atmosphere. (4). Measured maximum capillary pressure, kg/m^2 . (5). Calculated effective diameter, km . (6). Nominal diameter, μ . (7). Porosity, o/o. (8). Thickness of core. (9). Grid made of ~~the~~ stainless steel (80 thin wires ~~on~~^{per} 1 cm.). (10). methanol. (11). ~~pairs~~^{steam}. (12). The same. (13). air. (14). Grid ~~from~~^{of} bronze (80 thin wires ~~on~~^{per} 1 cm.). (15). Grid made of ~~the~~ stainless steel (80 wires ~~on~~^{per} 1 cm.). (16). gasoline. (17). water. (18). Grid ~~from~~^{of} bronze (80 thin wires ~~on~~^{per} 1 cm.). (19). Grid ~~from~~^{of} nickel (80 thin wires ~~on~~^{per} 1 cm.). (20). Porous copper (sintered from powder). (21). methanol. (22). Porous copper (sintered from shaving). (23). Porous nickel (sintered from powder). (24). Porous nickel (sintered from shaving). (25). Porous copper (sintered from shaving). (26). distilled water. (27). Porous copper oxidized (sintered from shaving). (28). the same. (29). Porous copper (sintered from shaving). (30). Porous copper oxidized. (31). Porous copper from ~~the~~ pressed nonsintered powder. (32). ~~is~~ Copper ~~net~~^{grid} (60 x 60 mesh size), wire size 0.165 mm. (33). Nickel ~~is~~ grid (50 x 50 mesh size), wire size 0.102 mm. (34). Nickel grid, ~~is~~ oxidized (50 x 50 mesh size), wire size 0.102 mm. (35). Nickel ~~is~~ grid (120 x 120 mesh size), wire size 0.076 mm. (36). The same, oxidized. (37). Glass cloth ASTT- (b) S-2 (10 layers). (38). Glass cloth of ~~5~~⁶-0.1 (30 layers). (39). Glass cloth ~~is the~~ nickel, fr. 0,2. (40). Glass cloth ~~of~~ ~~5~~⁶-0.1 (24 layers). (41). Sintered fiberglass ~~PS-1~~ with parallel laying. (42). Brass grid (four-celled, 23 layers). (43). acetone.

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The process of the determination of permeability consists of the following: is assigned the flow value of the liquid through the core, the core is inclined to horizontal to the angle ψ , in the chamber is ~~supported~~^{maintained} any determined pressure P , necessary for the formation of the determined radius of curvature of interface liquid, vapor. The flow value of liquid is varied until the pressure of liquid in core is establish^{as} ~~installed by~~ constant. Then is determined the flow value of liquid. Slope angle can vary within the range of 10 to 70° (for example when using as the working fluid ~~of~~ ethyl alcohol).

The pressure differential ~~on the~~^{per} unit of ~~the~~ length of core is equal to

$$\frac{\Delta P_m}{l} = \rho_m g \sin \psi = \frac{\gamma_m}{K} \cdot \frac{j_m}{S} \quad (2.25)$$

From this expression it is possible to determine the permeability of core. For fine/thin cores the determining factor is not so much the cross-sectional area of core, as its thickness C .

Table 3.

(1) Материал	(2) Пористость, %	(3) Размер волокна (частицы), мм	$K \cdot 10^{10}, \text{м}^2$
(4) Спеченная никелевая стружка	84	0,001	0,42
(4) Спеченная никелевая стружка	69	0,001	0,14
(4) Спеченная никелевая стружка	87	0,0015	0,30
(5) Спеченная стружка из нержавеющей стали	89	0,0030	5,3
(5) Спеченная стружка из нержавеющей стали	81	0,0030	1,9
(5) Спеченная стружка из нержавеющей стали	82	0,0080	11,3
(6) Спеченная пластина из порошка никеля	65	0,2	2,64
(6) Спеченная пластина из порошка никеля	54	0,3—0,6	0,78
(6) Спеченная пластина из порошка никеля	69	0,3—0,6	2,9
(7) Спеченная пластина из никелевых сеток	62	0,25	6,44
(7) Спеченная пластина из никелевых сеток	58	0,10	1,47
(7) Спеченная пластина из никелевых сеток	65	0,05	0,75
(8) Стеклоткань АСТП(Б)С-2 (10 слоев; 4,25 мм)	62	—	5
(9) Стеклоткань Э-0,1 (30 слоев, 3 мм)	60	—	0,39
(10) Пластина из спеченного никелевого порошка	33	0,2	2,6
(11) Латунная сетка	72	0,1	2,6

Key: (1). Material. (2). Porosity, o/o. (3). Size/~~dimension~~ of filament (particle), ~~of~~ mm. (4). Sintered nickel shaving. (5). Sintered shaving made of the stainless steel. (6). Sintered plate from the powder of nickel. (7). Sintered plate from nickel grids. (8). Glass cloth ASTP ⁽⁸⁾ S-2 (10 layers; 4.25 mm). (9). Glass cloth ~~of~~ ^{Э-0,1} (30 layers, 3 mm). (10). Plate from the sintered nickel powder. (11). Brass grid.

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The resistance of ~~fine~~^{is} thin porous cores usually defined as $R \sim 1/KC$:

$$R = \frac{\rho_m \sin \psi S}{v_{m0} j C} \quad (2.26)$$

Furthermore, supplementary resistance exerts the forces of the surface tension of interfaces liquid - vapor, which change as function of the length of core. Consequently, the resistance R is also the function of radius of curvature r .

Table 3 gives the results of experiments regarding the permeability of ~~the~~ different porous bodies of different thickness.

2. Use of low-temperature thermal ~~ducts~~ ^{tubes under} ~~in the mode~~/conditions of boiling.

Heat removal by boiling within the cores of the thermal ~~ducts~~ ^{tubes} and steam chambers up to now ^{has been} ~~is~~ studied insufficiently. Boiling liquid in porous cores begins as a result of the overheating of liquid relative to saturation temperature. In a ^(number of studies) ~~series~~ [10, 13, 15-17] is conducted the investigation of heat removal by boiling in thermal ~~ducts~~ ^{tubes} and ^(it is) shown that the ~~mode~~/conditions of ^{bubbling} ~~nucleate~~ boiling can occur in the normal state of the work of thermal ~~duct~~ ^{tube}. There are ^(reports) [86-89], where is negated the possibility of the functioning of thermal ~~ducts~~ ^{tubes under} ~~in the mode~~/conditions of boiling. In connection with this were carried out experimental studies on heat exchange by boiling liquid in the cores of thermal ~~ducts~~ ^{tubes} with the ^{feed} ~~supply~~ by ~~the~~ liquid of the zone of evaporator/~~vaporizer~~ with the aid of capillary forces.

Figure 13 shows ^a ~~the~~ diagram of experimental installation. Capillary-porous core 1 of fiberglass is wound around metallic pipe

40 cm. long ⁽ⁱⁿ⁾ several layers (7-15 layers). Inside of core at a distance ^{of} 87 mm from the upper edge of the ^{tube} ~~duct~~ between 4 and 5 layers of fiberglass was arranged heater ^(made) from Nichrome wire 0.2 mm in diameter. The resistance of wire was 25 Ω . To heater was connected ^a the source of direct current. Metal tube with the wound around it fiberglass core was placed ^{with} by butt end into container with liquid.

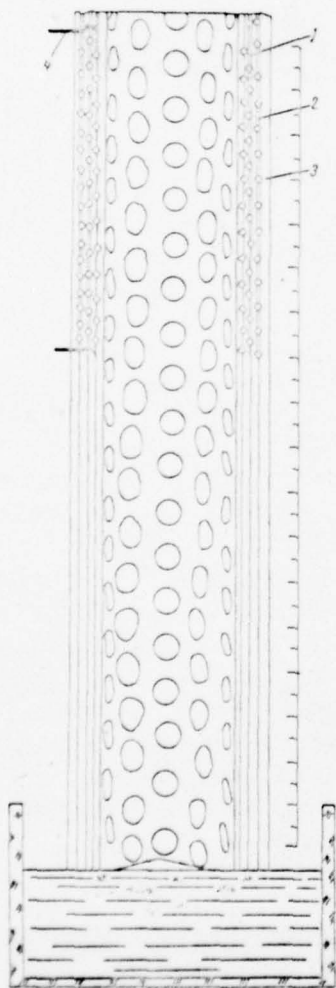


Fig. 13. Experimental installation ^{for} ~~on~~ the investigation of heat exchange in the cores of the thermal ^{tubes} ~~ducts~~: 1 - ~~the~~ perforated/~~punched~~ tube; 2 - heater; 3 - glass cloth; 4 - the ^{leads} ~~conclusion/derivations~~ of heater; 5 - container with liquid.

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With the absorption of liquid into core, was recorded the velocity of the motion of the front of the isopotential surface of liquid and the local values of the concentration of liquid with the aid of the electrical capacitance method of the measurement of concentrations and method of the dyeing/~~coloration~~ of the core with methyloange indicator. Furthermore, was recorded the ^{cubic}~~volume~~ velocity of the motion of liquid with the aid of the graduated glass container with divisions, from which occurred the ^{formation}~~makeup~~ by the liquid of bath 2 with the fixed level of ^{reflector}~~mirror~~.

In the first series of experiments, was investigated the kinetics of the absorption of liquid by capillary-porous core against the forces of gravitation with the ~~disconnection~~/cutoff of the source of heating. For this purpose were made several specimen/~~samples~~ of cylindrical capillary-porous cores made of fiberglass.

Specimen/~~sample~~ 1 was ^a skirt ~~from the~~ ^{made of} glass cloth, wound around ^{tube} ~~duct~~ made of the stainless steel (outer diameter 18 mm, thickness of ^{wall} ~~riding-crops~~ 0.2 mm) into 7 layers ^{with} and ^{of} the having outer diameter 19.3 mm. The Nichrome heater was ~~arrange~~/located between 4 and 5 layers of glass cloth at a distance ^{of} 120 mm from surface of liquid. The porosity of core was 600/o.

Specimen/~~sample~~ 2 had analogous form, but it was wound around ^{tube} ~~duct~~ 41 mm in diameter into 7 layers of glass cloth. The outer diameter of specimen/~~sample~~ ^{was} is equal to 47 mm. The heater was ~~arrange~~/located also between 4 and 5 layers ^{at} on height ^{of} 120 mm from surface of liquid. The porosity of core was 860/o.

Specimen/~~sample~~ 3 was wound around ^{tube} ~~duct~~ 18 mm in diameter into 15 layers of glass cloth. Heater was placed between 9 and 10 layers. The outer diameter of cylinder from glass cloth ^{was} is equal to 20.7 mm. The porosity of core was equal to 640/o.

As heat-transfer agent was selected ~~the~~ distilled water. The kinetics of the absorption of water into capillary-porous core from glass cloth was realize/accomplished in the atmosphere of saturated steam in order to eliminate the effect of the evaporation of liquid from the pores of core. In Fig. 14b is shown the curve of the kinetics of absorption. As can be seen from figure, experimental

points ^{adhere} ~~lie down~~ well to straight line $dh/dr = f(1/h)$.

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The maximum ^{height} ~~altitude~~ of the ^{raising} ~~elevation~~ of liquid for specimen/~~sample~~ 1 ^{was} ~~composed~~ $h_{\max} = 50$ cm., for the specimen/~~sample~~ of $2h_{\max} = 33$ cm., for the specimen/~~sample~~ of $3h_{\max} = 45$ cm. The linear speed of the motion of liquid in core at a distance, which corresponds ^{to} $h = 12$ cm. (center of the ^{placement} ~~arrangement~~ of heater), with ~~off~~ ~~heater~~ ^{was} ~~composed~~ $0.15 \cdot 10^{-3}$ m/s.

The distance of the center of heater from surface of liquid in container was selected in such a way that it would correspond to the height/~~altitude~~ of $h_{\max}/2$.

The second series of experiments on the absorption of liquid was realized/^{accomplished} as follows. Upon achieving steady state during the absorption of liquid into capillary-porous core (~~after~~ 6-7 hours after the beginning of experiment) to Nichrome heater was supplied the electrical ^{power} ~~energy~~ from ~~the~~ adjustable source of direct current.

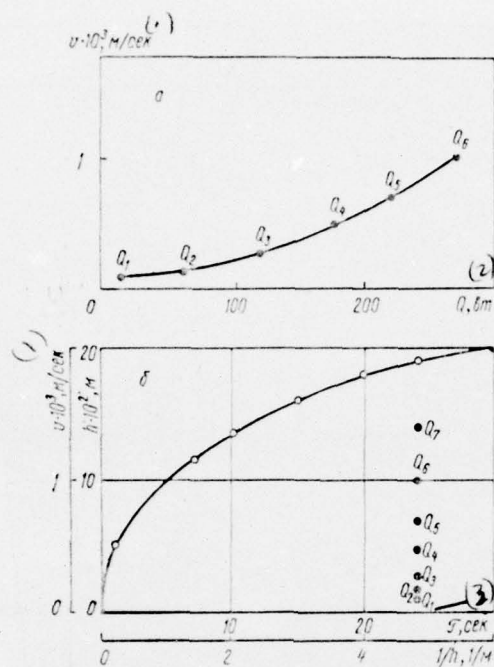


Fig. 14. Dependence of the rate of evaporation from core ^{on} from the applied power (a) and the kinetics of absorption (b).

Key: (1). m/s. (2). W. (3). τ , s.

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Under the action of Joule heat, occurred the evaporation or boiling of liquid ⁱⁿ on the section of the capillary-porous core, adjoining the heater. Was ~~realize~~/accomplished the measurement of the rate of evaporation of liquid and temperature field on the surface of porous core with the aid of copper-constantan thermocouples.

Fig. 15 ~~are shown~~ ^{at} ~~dependence~~ ^(of the dependence) curves of the temperature of the surface of core as functions of distance from the ^{surface} ~~mirror~~ of water and effective radius of meniscus ^{on} ~~from~~ rate of evaporation.

During investigation was noticed substantially ~~an~~ increase in the rate of evaporation with ^{the} ~~an~~ increase in power input Q (see Fig. 14a) for the glass cloth, which have the large size/~~dimension~~ of

cells, while for the glass cloth, which have a fine size/dimension of cells, ^{the} an increase in the velocity of absorption not as ~~it is~~ substantial. Probably this ~~it~~ is explained by the fact that the ~~yield~~ conditions ^(for entrance of steam into) pair in coarse-mesh glass cloth are more favorable in comparison with fine-mesh.

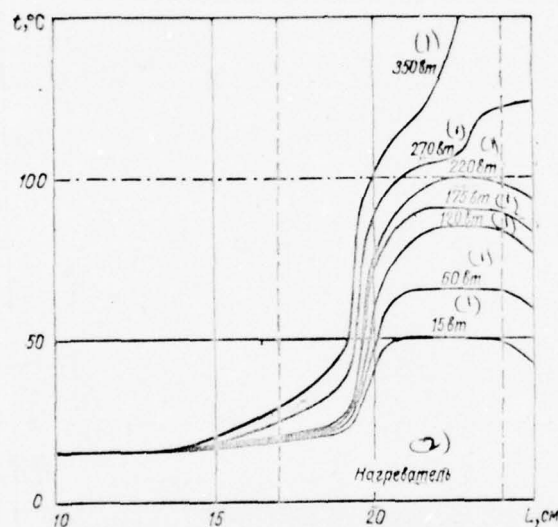


Fig. 15. Dependence of the temperature field of core on the applied power.

Key: (1). W. (2). heater.

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The removal/~~distance~~ of bubbles from porous material is accompanied by the local pulsations of pressure on boundary vapor - liquid. These local pulsations of pressure lead to the fact that, besides capillary potential and the potential of gravitational field, appears the ^{additional} ~~supplementary~~ driving/moving potential of the $\Psi_{\text{пульс}}$ which, ^{when} ~~while~~ averaged by the volume of evaporator/~~vaporizer~~, contributes to the absorption of liquid (effect, analogous to the work of diaphragm pump). In this case the generalized law of Darcy, in our opinion, can be ⁽²⁾presented in the form

$$j_m = -K(0) \text{grad } \Phi, \quad (2.27)$$

$$\Phi = \Psi_{\text{кап}} + zg + \Psi_{\text{пульс}}, \quad (2.28)$$

where ~~the~~ $\Psi_{\text{кап}} = f(A)$ - the potential of capillary absorption; z - the

coordinate ~~is~~, which characterizes the position of liquid in gravitational field; $\Psi_{\text{вытс}} = f(g)$ - the potential of the pulsating measurement of pressure in porous body in the presence of boiling or bubbling of the gas through pores.

The experiments on boiling liquid in porous body during the motion of liquid against gravitational forces with the aid of capillary forces and the forces of total pressure when the pulsations ^(steam) of bubbles are present, ~~pair~~ ^{significant} showed the ~~essential~~ influence of the latter for the porous bodies ^p having high permeability and ~~the~~ large size/dimension of pores.

2. For the explanation of the influence of porous structure on the process of boiling, were carried out the experiments on boiling ~~the~~ distilled water in ^a ~~the~~ porous core, consisting of several layers of glass cloth and arranged ^{located} horizontally. The distilled water was introduced directly inside porous core with the aid of several needles made of ~~the~~ stainless steel, connected ^{to graduated} with volumetric ^{cylinders} ~~glasses~~. Figure 16 shows installation diagram.

Installation is ~~the~~ vacuum-tight chamber of cylindrical form whose diameter is 470 mm and whose height is 670 mm. The chamber makes it possible to conduct research ^{on} of the process of boiling ^{of} both usual and cryogenic liquids, since it has ~~a~~ good vacuum thermal

insulation of the following construction: into chamber 1, is inserted vacuum-tight vessel 2. In the clearance between ~~the~~ chamber ~~of~~ 1 and 2 is maintained vacuum at 10^{-5} mm hg by means of diffusion mercury pump.

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Within vacuum chamber 2, is ^{located} ~~arranged~~ capillary-porous body 3 with the feed system of liquid, the system of the heating of body and condensation ^{tube} ~~duct~~ for condensation and collection of the vapors 14. Thus, within chamber 2 is realize^d ~~accomplished~~ ^a ~~the~~ closed loop of ~~the~~ evaporation - condensation with the return of the condensed liquid to the place of evaporation into container 7, whence liquid proceeds to capillary-porous body through core.

Recording ^{of} temperature fields was realize^d ~~accomplished~~ with the aid of thermocouples and 24- point electronic potentiometer. The inspection of the correctness of readings of thermocouples was realize^d ~~accomplished~~ by a potentiometer of R-306 .

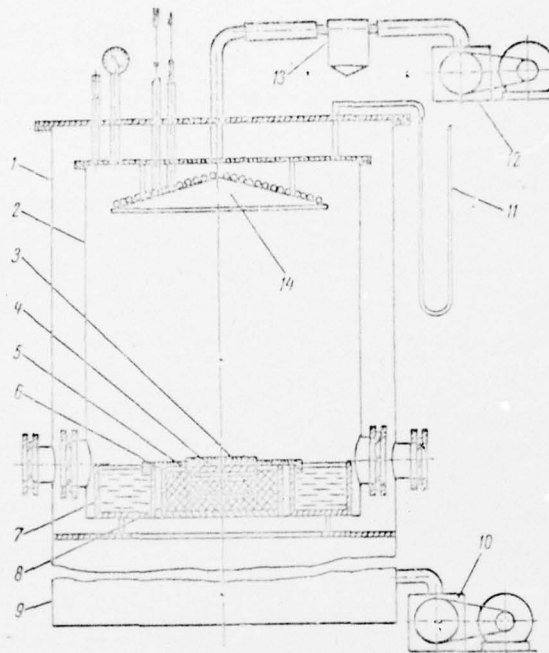


Fig. 16. Experimental installation for the study of evaporation from capillary-porous body; 1 - ~~the~~ basic volume; 2 - vacuum volume; 3 - the investigated capillary-porous body; 4 - copper plate; 5 - heater; 6 - adiabatic ^{shell} enclosure; 7 - the investigated liquid; 8 - thermal insulation; 9 - diffusion pump; 10 - fore pump; 11 - U-tube gauge; 12 - mechanical vacuum pump; 13 - trap; 14 - conditional ^{ing system} duct.

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Pressure within vacuum chamber 2 was measured with the aid of ~~the~~ ^a system of ~~the~~ U-tube gauges, filled ~~by~~ ^{with} mercury, by the organosilicon liquid ~~VKS~~ ^{VKS and} by dibutyl phthalate. The removal/~~distance~~ of non-condensable gases from chamber 2 was ~~realize~~ ^{realize}/accomplished by mechanical pump VN-2MG (12). For the ~~destruction by frost~~ ^{freezing} of the vapors of liquid, caught ~~into~~ the system of suction, was utilized nitrogen trap 13.

Porous body consisted of ~~the~~ different amount of layers of glass cloth; it had a length ^{of} 150 mm and a width 50 mm. It was pressed by ~~framework/body~~ against the heating plate, ~~arrange~~ ^{located} on the external surface of experimental box.

The system of heating had two lamellar electric heaters - ~~basic~~ ^{main}

and ^{reserve} ~~guard~~ (adiabatic ^{shell} ~~enclosure~~). Heaters were asbestos-cement framework/bodies with ~~the~~ grooves, into which was placed ~~the~~ wire ^{of} Kanthal (specific resistance of Kanthal ^{is} 1.45 ohm/mm³, maximum operating temperatures ^{is} 1375°C). Above the ^{main} ~~basic~~ heater was ~~arrange~~/located ^a ~~the~~ copper plate on which was placed capillary-porous body. The electrical insulator between the plate and spiral ^{of} ~~the~~ heater was ^a ~~the~~ thin plate of mica.

Heat flux was regulated by autotransformer and was recorded with the aid of ammeter and voltmeter of class of accuracy 0.5. Between the ^{main} ~~basic~~ and ^{reserve} ~~guard~~ heaters in the process of experiments, ^{maintained} ~~supported the~~ zero gradient of temperature for the creation of the adiabatic conditions of heating. The heat flux, created by this system of heating, could reach value 12 W/cm². Experiments were carried out during a change in the heat flux from 0.5 to 11 W/cm².

The ~~expenditure~~/consumption of liquid per unit time with ^a ~~the~~ fixed heat flux was determined as follows:

$$I_{\text{ж}} = \frac{qA}{r'}, \quad (2.29)$$

where ~~the~~ j_m - the fluid flow rate per unit time; q is ~~a~~ heat-flux density; A - the surface area of heating; r' - heat of vaporization.

The ^{rate of} ~~speed control~~ the supply of liquid was ^{controlled} ~~realize/accomplished~~ with the aid of ~~throttle~~/choke by changing the flow area of connecting tubes.

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By this control it was possible to ensure ^a ~~the~~ feed rate of liquid, equal to rate of evaporation.

For the measurement of temperature field, ~~they~~ were utilized six chromel-aluminum thermocouples. One of them was caked ~~into~~ the copper plate, ^{while} the others were ~~arrange~~/located in capillary-porous body.

As capillary-porous bodies were ^{used} ~~applied~~ the mill packs of the glass cloth of two forms: a) ASTT-b-S-2 MRTU [MPTY - Interrepublic Technical Specifications] 6-11-140-70 with a minimum radius of ~~the~~ pores of $r_{min} = 53.7 \mu$, permeability $K = 5 \cdot 10^{-6} \text{ cm}^2$; b) ~~E-0.1~~ GOST [ГОСТ - All-union State Standard] 8481-61 with a minimum radius of ~~the~~ pores of $r_{min} = 52.4 \mu$, permeability $K = 2.7 \cdot 10^{-7} \text{ cm}^2$.

In conducting the experiments ^a on heat removal by boiling in ~~the~~ porous core ^{made} from ~~the~~ package of glass cloth, arranged ^{located} ~~it is~~ horizontal, ^{by} was recorded the heat flux q , temperature field over the ^{cross} section of porous body and the fluid flow ~~of~~ j_m . Moreover, the fluid flow ~~of~~ j_m was selected so that ^{total} ~~whole~~ amount of heat, isolated by heater, would go to the evaporation of liquid

$$q = j_m r'. \quad (2.30)$$

On the other hand,

$$q = \alpha (T_{cr} - T_{nac}), \quad (2.31)$$

where α is ^{the} coefficient of heat exchange.

Figure 17a, b depicts the curve ^{graphs} ~~graphs~~ of dependence of q on ΔT and α on q for ~~the~~ ^{types} different ~~forms~~ of porous cores.

The most favorable conditions of heat exchange ^{by} ~~boiling~~ proved to be ⁱⁿ ~~at~~ glass cloth 1 since it had the high value of permeability k as compared with glass cloth 2. The crisis of boiling substantially ~~was shift~~ ^{shifted} ~~sheared~~ to the side of large heat fluxes in the presence of

grooves in porous core, which contributed to diversion ^{of steam} ~~tap pair~~.

The tests showed that ~~the~~ basic influence on heat exchange during boiling exerts the conditions of diversion ^{of steam} ~~tap pair~~ from capillary-porous body. By comparing the values of the coefficient of heat exchange ^{of} glass cloths 1 and 2, it is possible to note following: ⁱⁿ glass cloth 2, maximum value of the coefficient α above and during the ^{easing} ~~facilitation~~ of the conditions of diversion ^{of steam} ~~tap pair~~ increases and is displaced to the side of large heat fluxes faster than ⁱⁿ glass cloth 2. Here, it is obvious ^{that} large role ^{is} played ^{by} the permeability of body K, which is proportional to the root-mean-square size ~~dimension~~ of pores.

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^{In} glass cloth 1 permeability K ^{is} 18 times ^{greater} ~~is more~~ than ⁱⁿ glass cloth 2. This causes an improvement in ~~the~~ heat exchange during boiling because of the facilitation of diversion ^{of steam} ~~tap pair~~ ^{by} on the pores of larger diameter, but ⁱⁿ on the grid/~~network~~ of capillaries with small size/~~dimensions~~ water ~~it~~ continues to ^{reach} ~~enter~~ under the action of capillary forces, the heat surface.

An improvement in the conditions of steam discharge pipe because of the formation of ~~the~~ steam-removal channels, as ~~it~~ takes place in

the case with glass cloth 1, and the facilitation of diversion ~~from~~ from sides, as ~~it~~ takes place with glass cloth 2, ~~it~~ leads to an increase in the coefficient of heat exchange, the displacement of its maximum to the side of great ^{value} ~~significance~~ of heat flux and decrease in the overheating of wall. Is reached an increase in the value of ^{tical} ~~critical~~ heat flux.

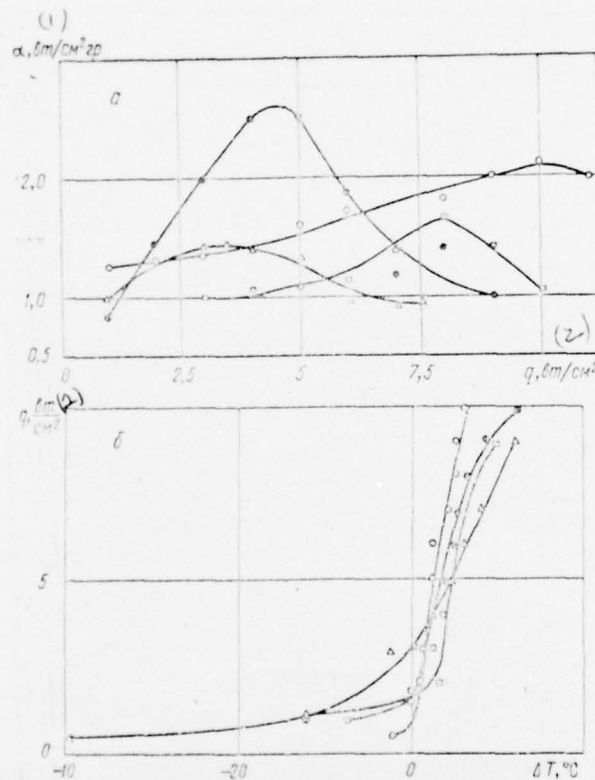


Fig. 17. Dependence of the coefficient of heat exchange α of boiling on the applied power (a) and the applied power on the temperature differential in porous core (b).

Key: (1). $W/cm^2 \cdot deg$. (2) W/cm^2 .

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In the case of a decrease in the thickness of capillary-porous body, in essence, also is observed an improvement in ~~the~~ heat exchange; however, fine/~~thin~~ capillary-porous body badly/~~poorly~~ organizes the process of boiling. It is unstable, occurs the wavy ejection of liquid from body, accompanied by sharp sounds, by the distention of body, ^(and) by temperature jumps in capillary-porous body.

Here, it is obvious, large role ^{is} played ^{by} the following torque/moments. On one hand, ⁱⁿ of fine/~~thin~~ capillary-porous body is ^{facilitated} ~~low-duty~~ the output/^{steam} ~~yield pair~~ because of ^{the} ~~a~~ decrease in the path of its passage in body, which favorably affecting ^{the} ~~the~~ heat exchange. On the other hand, the ^{fine} ~~slender~~ body had less volume, ^{so} ~~but~~ therefore ^a ~~the~~ larger percentage of its pores was filled by liquid (flow rate of applied liquid depended on heat flux). During boiling, ^{of} ~~vapor~~ intensely ^(was) ~~discarded~~ liquid of the pores of larger size/~~dimension~~.

Curve $\alpha = f(q)$ for glass cloth 2 have ^{the} ~~a~~ form ~~of form~~

$$y = Bx^m \exp Cx.$$

(2.32)

In particular for ~~a~~ glass cloth with ~~a~~ thickness 3.6 mm (36 layers) this formula is revealed as follows:

$$y = (1,25 + x) x^{3,75} \exp(-1,5x) \quad (2.33)$$

or

$$\alpha = (1,25 + q) q^{3,75} \exp(-1,5q). \quad (2.34)$$

It is obvious, ^{that} for ~~a~~ glass cloth ^{1,} dependence of the same form, but limitation along heat flux (to 11 W/cm²) in experiments ~~they~~ did not make it give possible ^{up} ~~to end/lead~~ to explain form ^{of} curve $\alpha = f(q)$.

Boiling as process ^{as} extremely complex gives the very large spread of experimental data. ~~That it is~~ ^{All the} more in experimental datum, when the number of factors, which determine heat exchange, grows ^{because} ~~raises~~ because of the presence of capillary-porous body. It is hence difficult to propose any formulas for the calculation of heat exchange. However it is possible to ~~state~~ establish that an improvement in the conditions of diversion ^{of steam} ~~tap the pair~~ leads to the intensity of heat exchange. Dependence $\alpha = f(q)$ is represented by expression

$$\alpha = B_1 q^m \exp Cq. \quad (2.35)$$

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The structure of capillary-porous body ^{determines} ~~defines~~ the type of the motion of ^{steam} bubbles ~~pair~~ and the place of the origin/~~conception/initiation~~ of bubbles. So, ^{the} ~~a~~ structure of the type of the package of grids or fabrics has ^{greater} permeability K along filaments ~~greater~~ than across the grain. Therefore ^{steam} bubbles ~~pair~~ attempt to ~~be~~ moved easier along filaments than across the grain. In connection with this, the grooves and the steam-removal channels can substantially improve heat exchange by boiling, since ^(they) facilitate the conditions for ^{of steam} ~~a~~ output/~~yield~~ ~~pair~~.

In this series of experiments, ^{steam} was revealed the role of the pulsations of pressure upon the entrance of bubbles ~~the pair~~ from porous body ⁱⁿ ~~to~~ the process of ~~the~~ wetting with the liquid of capillary-porous body. Since the liquid was introduced into the porous body through needles at isolated points, when only capillary forces of transfer are present, we would ^{reach} ~~achieve~~ limit ^{of} ~~on~~ the transport of fluid flow much earlier than ^{the case} ~~this was~~ in experiments. Furthermore, during the unidirectional motion of liquid by means of capillary forces ^{we would have} ~~in us~~ the better/~~best~~ conditions of cooling ~~would be~~ ^{at} in the beginning of evaporator/~~vaporizer~~ and worse ^{at} ~~in~~ its end/~~heat~~.

Readings of thermocouples, however, indicate that the

temperature field within entire porous body was uniform. Hence it follows that the supply by a liquid was realized ~~accomplished~~ evenly ^{over} ~~by~~ entire volume. This could be realized only with the aid of the supplementary mechanism of the transfer of liquid.

Let us attempt to make ^{an} analysis of the influence of two-phase flow ^{on} ~~to~~ the process of the capillary absorption of liquid in porous body. In the presence of the process of boiling in porous body, occurs two-phase fluid flow - ^{steam} ~~pairs~~. Since the thermal conductivity of porous core in low-temperature thermal ^{tubes} ~~ducts~~ usually ^{is} several times ~~is~~ higher than the thermal conductivity of liquid, are created the favorable conditions for the origin ^{ation} ~~conception~~ / initiation of ^{steam} ~~of~~ bubbles ~~part~~ and it is possible to ^{say} ~~conclude~~ that in the zone of evaporation occurs boiling ^{of} liquid under conditions of saturation. In work [90] ~~it~~ is discussed ~~about~~ the fact that for the process of the motion of two-phase flow also it is possible to utilize ~~a~~ law of Darcy .

For a liquid

$$j_m = -K \frac{K_1 \rho_m}{\mu_m} (\text{grad } P_1 - \rho_m g \sin \varphi) A, \quad (2.36)$$

for ~~pair~~ steam

$$j_n = -K \frac{K_2 \rho_n}{\mu_n} (\text{grad } P_2 - \rho_n g \sin \varphi) A, \quad (2.37)$$

$$\Pi \frac{\partial (\rho_n S_1)}{\partial \tau} = -\text{div} \left(\frac{j_n}{A} \right), \quad (2.38)$$

$$\Pi \frac{\partial (\rho_n S_2)}{\partial \tau} = -\text{div} (j_n/A), \quad (2.39)$$

$$S_1 + S_2 = 1, \quad \rho_m = \text{const}, \quad \rho_n = \text{const}, \quad (2.40)$$

$$P_2 - P_1 = \Delta P_{\text{cap}}(S_1),$$

where K is ^{total} ~~general~~ permeability; $K_{1,2}$ - relative permeability (in fractions of the ~~general/general~~/total); S is ~~a~~ concentration of liquid; Π - porosity; ΔP_{cap} - capillary pressure.

The relative permeability K_1 and K_2 are only functions of the

concentration of liquid.

If one assumes that ^{all the} ~~whole~~ heat in evaporator/~~vaporizer~~ goes to heat of vaporization, then under the stationary equilibrium conditions ~~of~~ $l_m = l_n$; therefore, the law of Darcy can be written in general form

$$j = K \frac{\rho_m A}{\mu_m} (\text{grad } P + g \rho \sin \alpha), \quad (2.41)$$

$$j = \frac{Q}{r'} = \frac{qA}{r'}. \quad (2.42)$$

Here $\text{grad } p$ is ~~a~~ pressure gradient in the porous medium due to the presence of forces of friction;

$$\frac{dP}{dy} = - \left[\frac{\mu_m}{K} \frac{q}{r' \rho_m} + \rho_m g \sin \varphi \right], \quad (2.43)$$

$$\Delta P_{\text{tp}} = -L \left[\frac{\mu_m q}{K r' \rho_m} + \rho_m g \sin \varphi \right], \quad (2.44)$$

$$\Delta P_{\text{cap}} = \frac{2\sigma \cos \theta}{R_{\text{cp}}}, \quad (2.45)$$

where the R_{cp} - the mean radius of pore.

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If in porous body there are ^{steam} bubbles ~~pair~~, then the pressure which attempts to expand bubble, is equal to

$$P_{nap} = \frac{2\sigma}{R_{cp}}. \quad (2.46)$$

Let us assume that the process of heat transfer from the heated wall to the surface of core is realized ~~accomplished~~ by means of thermal conductivity through the film of vapor with ~~a~~ thickness δ_1 and then fluid film $C - \delta_1$. Heat flux through the fluid film is equal to

$$q = \lambda_m \frac{T_{cr} - T}{C - \delta_1}. \quad (2.47)$$

The pressure, which attempts to ^(press) bubble, is equal to

$$P_{na} = P_{osm} + \frac{2\sigma}{r_n}, \quad (2.48)$$

where r_n - the radius of bubble.

Pressure in ^{steam} bubble ~~air~~ and ^(in fluid with) fluid pressure ~~by~~ which is saturated the porous core, must be equal to each other. Since the pressure of liquid is equal to pressure ^(of steam) ~~in~~ within bubbles, it must be more than the saturation pressure ~~of~~ P_{nac} in the steam space of thermal ^{tube} ~~duct~~.

$$P_m = P_n + \frac{2\sigma}{\rho}. \quad (2.49)$$

According to the conditions of heat exchange, in the layer of liquid there is a gradient of temperature. In accordance with the equation of Clapeyron - Mendeleyev

$$\Delta P_{nap} = \frac{P_{nac} r'}{RT_{nac}^2} \Delta T. \quad (2.50)$$

This formula gives a pressure difference between ~~of~~ $P_{\text{нас}}$ and $P_{\text{ж}}$ as a result of the presence of the overheating of liquid relative to T_{cr} .

Thus, the balance of face pressure ^{on liquid-steam interface} ~~of section liquid~~ pairs

$$P_{\text{пар}} - P_{\text{нас}} = \Delta P_{\text{кан}} + \Delta P_{\text{пар}}. \quad (2.51)$$

Substituting (2.44), (2.45), (2.50) and (2.51) into equation (2.48), we obtain the thickness of ^{steam} film ~~pair~~ in porous core depending on heat-flux density

$$\delta_1 = C - \frac{\lambda_{\text{ж}} R T_{\text{нас}}^2}{q P_{\text{нас}} r'} \left[\frac{2\sigma \cos \theta}{R_{\text{ep}}} - \rho_{\text{ж}} L \sin \alpha - \frac{q L_{\text{ж}} L \mu_{\text{ж}}}{r' \rho_{\text{ж}} K C} \right]. \quad (2.52)$$

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This equation is correct, when $\delta_1 < C$. Negative δ_1 indicates that the capillary pressure flattened cushion ~~pair~~.
steam

On the basis entire stated above ^(information) it is possible to make the conclusion that the low-temperature thermal ^{tubes} ~~ducts~~ can work in the mode/conditions of ^{bubbling} ~~nucleate~~ boiling; however, in this case ~~grow~~ rises the thermal resistance of porous core, ^{which} ~~that it~~ leads to an increase in the temperature differentials ΔT along ^{tube} ~~duct~~. An increase in the thermal resistance of core appears because in evaporator/vaporizer the liquid is ~~located~~ in two-phase state (bubbles ^{of steam} ~~pair~~ - liquid). ~~of~~ the Pulsation of the pressure of liquid ^{during exit} ~~on leaving~~ of (bubbles ^{steam} ~~pair~~) contributes to the wetting of porous core, which is equivalent to an increase in the capillary pressure. The critical value of the overheating of liquid in porous core approximately can be ~~is~~ evaluated according to formula [15]

$$T_{cr} - T_{nac} = \frac{2\sigma T_{nac}}{r' \rho_n r_n}, \quad (2.53)$$

where r_n - ~~a~~ critical radius of ^{steam} bubble pair.

The heat exchange in the ~~mode~~ conditions of boiling ^{of} liquid in porous body is determined from formula [91]

$$\left(\frac{\alpha}{C_{ж/ж}} \right) \left(\frac{C_{ж/ж} \mu_{ж}}{\lambda_{ж}} \right) \left(\frac{\rho_{ж} \sigma}{P_{нас}^2} \right)^{0.21} = 0.072 \left(\frac{L_{н/ж}}{\mu_{ж}} \right)^{-0.77}, \quad (2.54)$$

where

$$j_{ж} = \frac{Q}{S \pi r'}.$$

The experimental data indicate ~~that~~ α during boiling ^{of} liquid in porous body with small heat fluxes can 2-3 times exceed α during boiling ^{of} liquid ^{with} ~~by~~ large volume.

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Chapter 3.

EXPERIMENTAL STUDY OF THERMAL ^{Tubes}~~DUCTS~~ AND THEIR APPLICATION ~~IN~~ IN
DIFFERENT BRANCHES OF INDUSTRY.

1. Cooling and the thermostatic control ^{of} semiconductor devices with
the aid of ~~the~~ thermal ^{tubes}~~ducts~~ and ~~the~~ steam chambers.

The thermal ^{tubes}~~ducts~~ and ~~the~~ steam chambers proved to be very
convenient ^{device for cooling}~~chillers~~ and thermostatic control of semiconductor

devices. Their use as the radiators of semiconductor devices made it possible significantly to improve the conditions of cooling and thermostatic control of the latter, and consequently, to improve their operational characteristics.

The diverse variants of the connection of thermal tubes to semiconductor devices are examined in works [25, 26]. In the described [25, 26] versions of the connection of thermal ~~tubes~~^{tubes} to transistors the assembly of semiconductor devices ~~is~~ is ~~realize~~/accomplished on the external wall of thermal tubes. The transmission of thermal energy from fuel element to thermal ~~tube~~^{tube} is ~~realize~~/accomplished through contact resistance^{of} the housing of instrument - the wall of thermal ~~tube~~^{tube}. Even if we for a decrease in ~~the~~ contact thermal resistance utilize ~~the~~ special lubrication, which possesses high thermal conductivity, then also in this case the temperature differential during thermal contact resistance will compose 0.4-1.5 deg/cm² ~~of~~^{of} surface^{of} at passage^{of} 1 W of heat output. Thermal ~~tube~~^{tube} as radiator cannot decrease this ~~the~~ temperature differential.

Thus, the connection of thermal ~~tube~~^{tube} to the semiconductor device or any other heat-releasing object can be considered as improvement of radiator.

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Heat withdrawal from semiconductor device to thermal ~~duct~~^{tube} radiator takes place: by heat withdrawal by thermal conductivity through the crystal of the instrument: by heat withdrawal through thermal contact resistance ^{of} crystal - ~~the~~ housing of transistor; by heat removal by thermal conductivity through the material of the housing of transistor; by heat withdrawal through contact resistance ^{of} the housing of transistor - the wall of radiator (thermal ~~duct~~^{tube}).

This can be explained by block diagram in Fig. 18a, where ~~the~~
 $R_{кр}$ ^{is} ~~are~~ the thermal resistance of the crystal of semiconductor;
 $R_{корп}$ - the thermal resistance of the housing of transistor;
 $R_{к1}$ - the resistance of contact ~~the~~ crystal ~~of the case~~^{housing} of transistor;
 $R_{к2}$ - contact resistance ^{of} the housing of transistor - radiator;
 $R_{рад}$ ~~are~~ the thermal resistance of the wall of radiator (thermal ~~duct~~^{tube}).

Thus, the mechanical connection of semiconductor device with thermal ~~duct~~^{tube} makes it possible to decrease ~~the~~ only thermal resistance of the radiator ~~of~~ $R_{рад}$ - leaving all the remaining thermal resistance ^{as} previous. This especially is aggravated in the work of transistors in vacuum.

In this work is proposed the method of the intensification of the process of heat exchange between the heat-releasing element (semiconductor device) and thermal ~~chamber~~ ^{tube} or steam chamber [27]. Its essence lies in the fact that the fuel element is introduced directly inside thermal ~~chamber~~ ^{tube} and is enveloped by capillary-porous core (Fig. 19c).

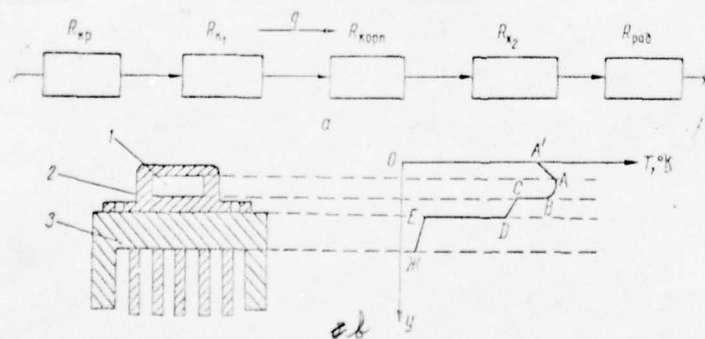


Fig. 18.

Fig. 18. The block diagram of heat withdrawal from semiconductor device (a); the diagram of semiconductor diode (b): 1 - radiator; 2 - housing; 3 - crystal; the distribution of temperature (c).

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The intensification of the process of heat exchange is achieved by the fact that cooling ^{of} fuel element is realized ~~accomplished~~ by evaporation or boiling ^{of} liquid in the pores of the capillary-porous core, which envelops fuel element. Cooling ^{of} semiconductor device or another source during its location inside the core of thermal ~~tube~~ ^{tube} makes it possible to feed liquid directly to the wall of the object of heat release, to distribute it evenly ^{over} ~~by~~ its entire surface with the aid of capillary forces. This method of cooling eliminates a series of thermal resistance^s in diagram (see Fig. 18a), in particular thermal contact resistance ~~the~~ wall - wall.

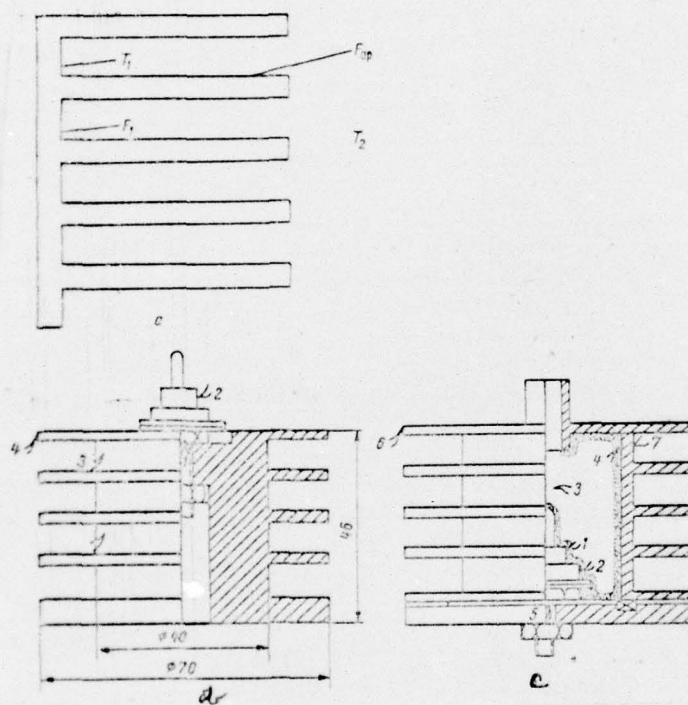


Fig. 19.

Fig. 19. Different forms of ~~the~~ radiators: a). the simplified form of ribbing; b). monolithic radiator; c). hollow radiator with semiconductor device inside, ~~the~~ covered ^{by} porous core.

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The supply of liquid to the object of heat release with the aid of capillary forces makes it possible to successfully cool it both in the gravitational field and in the presence of vibration and accelerations. The evaporation of liquid in the pores of capillary coating in the absence of non-condensable gas makes it possible to increase the coefficient of heat exchange as compared with the method of cooling, described in [25, 26].

Analogously the location of fuel element inside porous core positively ~~promotes also~~ ^{influences} the conditions of heat removal by boiling, after stabilizing this process.

The arrangement of the covered with porous material semiconductor devices within the steam chambers permits implementation of a construction of ~~the~~ airtight boxes, filled ^{with} ~~by~~ radio-electronics equipment, having prolonged guaranteed life both in gaseous or liquid medium and in vacuum. Probably the use of individual thermal ~~units~~ ^{tubes}

for each semiconductor device or a micromodule ^{is} not always possible, too high a density of assembly in contemporary installations and apparatuses. Technologically ^{it is} much simpler cover ^{the plates} ~~heat exchangers~~ with the placed on them sources of heat release (for example BIS) with porous dielectric coating, for example, pressing on ^{of} fiberglass grids on ^{plates} ~~heat exchangers~~ and to place them into the steam chamber, partially filled with dielectric heat-transfer agent, for example ~~by~~ Freon, in the pores of core. Such boxes can have standard detachable joints ^{and} ~~also~~ with their aid it is possible to rapidly ^{assemble} ~~combine~~ the necessary switching circuit. The location of semiconductor devices inside the airtight steam chamber or the thermal ^{tube} ~~duct~~ makes it possible to remove their metallic housing and to bare the crystals of semiconductors. This measure also will improve the conditions of their cooling. After the publication ^{of} [27] in the American journal "Electronics" [25] it was shown ¹ ~~the fact~~ that the analogous method of cooling ^{of} semiconductor devices begins to ^{be} utilize ^{by} the firm "TRW Systems" (Redondo Beach, California) for cooling power transistors.

Let us examine the method of ~~the~~ calculation of the intensity of heat withdrawal from the semiconductor device, placed into ^a ~~the~~ steam chamber.

The amount of heat, ~~abstracted~~ removed from the semiconductor device, placed into the steam chamber, and also its geometric dimensions and the area of capacitor it is possible to calculate from formulas (1.136), (1.137).

It is possible to consider that in the evaporator ~~vaporizer~~ of the steam chamber or thermal ~~core~~ ^{tube} occur the boundary conditions of the 2nd kind ~~of~~ ($q_w = \text{const}$). In this case, is absent the thermal resistance of chamber wall, since the object of heat release is located directly within core. On the external surface of capacitor, there can be different boundary conditions. From them most typical are the boundary conditions of the 2nd kind ~~of~~ ($q_w = \text{const}$), although can be encountered also the boundary conditions of the 3rd kind ($\alpha = \text{const}$) or of the 1st kind ~~of~~ ($T_w = \text{const}$) ..

If porous core takes the form of plate with the ~~size~~ dimensions of $L = (L_w + L_{w,3} + L_w)$; b and c (Fig. 20), then with the assigned heat flux in the zone of condensation q_w (boundary conditions of the 2nd kind) the maximum length of capacitor is equal to

$$L_{\text{max}} = 2 \sqrt{\frac{\rho_w r' \sigma}{\mu_w} \cdot \frac{c}{q_{\text{wmax}} K_1 R_{\text{min}}}} \quad (3.1)$$

~~and~~ and with the assigned length ~~of~~ L_n ^{t_n} heat flux in the zone of condensation

$$q_n = -K_{1n} \mu_n r' \Pi^2 c - \sqrt{K_{1n}^2 \mu_n^2 r'^2 \Pi^4 c^2 + \frac{4r'^2 \rho_n \Pi^2 c^2 \sigma}{L_n^2 R_{min}}} \quad (3.2)$$

But since the amount of heat, isolated in evaporator ~~is equal to~~, is equal to the amount of heat, ~~removed~~ removed in capacitor, ^{then}

$$q_n S_n = q_n L_n b, \quad (3.3)$$

hence

$$q_n = q_n \frac{L_n b}{S_n},$$

where S_n - the surface area of the core above the semiconductor device.

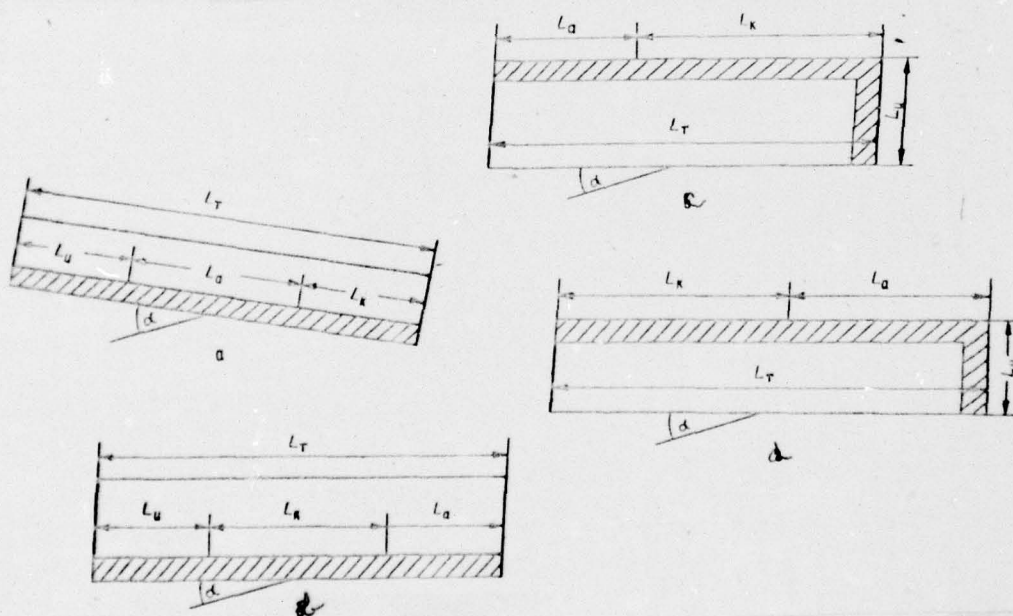


Fig. 20.

Fig. 20. Versions of the arrangement of the zones of ~~the~~ evaporation:

- a). evaporator ~~vaporizer~~ is located above, below it adjoins ~~capacitor~~ ^{condenser}; b). adiabatic zone is arranged between the evaporator ~~vaporizer~~ and the ~~capacitor~~ ^{condenser}; c). ~~capacitor~~ ^{condenser} and evaporator ~~vaporizer~~ are located on mutually perpendicular planes and adjoin each other; d). adiabatic zone is arranged between the evaporator ~~vaporizer~~ and the ~~capacitor~~ ^{condenser}.

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If the steam chamber has a form of ^a ~~the~~ cylinder, part of the lateral surface ~~of~~ $2\pi r_l L_k$ of which is ~~capacitor~~ ^{condenser}, ~~and~~ ^{also} on which is assigned ~~prescribed~~ the constant heat flux of q_k , ~~then~~

$$L_{\text{max}} = \left[\frac{\sigma/R_{\text{min}}}{\frac{\pi^2}{\rho_{\text{ж}} \Gamma^2} + K_1 \frac{\mu_{\text{ж}}}{\rho_{\text{ж}}} \cdot \frac{\pi}{2}} \right]^{1/2}, \quad (3.4)$$

where

$$\pi = \frac{q_k r_l}{r' (r_l^2 - r_0^2)}.$$

The maximum amount of heat, scattered by ~~capacitor~~ ^{condenser}, is equal to

$$Q_{\max} = -m - \left(\frac{m^2}{2} - 4\pi^2 \Pi^2 (r_l^2 - r_0^2)^2 \frac{\rho_{\text{ж}} \sigma r'^2}{R_{\min}} \right)^{1/2}, \quad (3.5)$$

where

$$m = r' K_1 \Pi^2 \frac{L_{\text{ж}}}{2} \mu_{\text{ж}} (r_l^2 - r_0^2).$$

Since the sources of heat release within the steam chambers can be arranged ~~located~~ in different places, let us examine several versions of the location of the zones of evaporation, condensation and adiabatic zone.

In gravitational field one should examine two characteristic cases of the arrangement of these zones, if evaporator ~~and condenser~~, condenser and adiabatic zone are located in one plane and are united by porous core in the form of plate.

Case 1. Evaporator ~~and condenser~~ is located above, below it adjoins ^{condenser} ~~capacitor~~ and adiabatic zone (Fig. 20b). The angle of the slope of core to line of horizon is equal to α :

$$q_{\text{max}} = \left[\left(\frac{2\rho_{\text{ж}} r' \sigma}{\mu_{\text{ж}}} \right) \left(\frac{\rho_{\text{ж}}}{g\sigma} \right) \left(\frac{c}{L_{\text{a},3}(L_{\text{ж}} + L_{\text{н}})} \right) - \right. \\ \left. - \frac{2\rho_{\text{ж}}^2 r'}{\mu_{\text{ж}}} \left(\frac{c}{L_{\text{a},3}(L_{\text{ж}} + L_{\text{н}})} \right) \sin \varphi \right] \frac{h_{\text{max}}}{K}. \quad (3.6)$$

Case 2. Adiabatic zone is arranged between the evaporator ~~evaporiser~~ and the ~~capacitor~~ ^{condenser}, whereupon evaporator ~~evaporiser~~ is arranged above (Fig. 20a)

$$q_{\text{max}} = \left[\frac{2\rho_{\text{ж}}^2 r'}{\mu_{\text{ж}}} \left(\frac{c}{L_{\text{н}}(L_{\text{н}} + L_{\text{к}}) + 2L_{\text{н}}L_{\text{а.з}}} \right) - \frac{2\rho_{\text{ж}}^2 r'}{\mu_{\text{ж}} h_{\text{max}}} \left(\frac{c(L_{\text{н}} + L_{\text{н}} + L_{\text{а.з}})}{L_{\text{н}}(L_{\text{н}} + L_{\text{к}}) + 2L_{\text{н}}L_{\text{а.з}}} \right) \sin \varphi \right] \frac{h_{\text{max}}}{K} . \quad (3.7)$$

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Frequently the evaporator ~~evaporizer~~ and ^{condenser} ~~capacitor~~ can be arranged ~~located~~ on interperpendicular planes, the adiabatic zone can be arranged ~~located~~ either about ^{condenser} ~~capacitor~~ from the side, opposite from evaporator ~~evaporizer~~ or between them.

1. The ^{condenser} ~~capacitor~~ and the evaporator ~~evaporizer~~ are arranged ~~located~~ on interperpendicular planes and adjoin each other. Adiabatic zone is located from opposite side ^{of condenser} ~~capacitor~~ (Figs. 20c), $0 \leq \phi \leq 90^\circ$:

$$q_{\text{ymax}} = \left[\frac{2\rho_{\text{ж}}^2 r'}{\mu_{\text{ж}}} \left(\frac{c}{L_{\text{н}}(L_{\text{н}} + 2X_{R_{\text{min}}}) - X_{R_{\text{min}}}^2} \right) - \right. \\ \left. - \frac{2\rho_{\text{ж}}^2 r'}{\mu_{\text{ж}}} \cdot \frac{1}{h_{\text{max}}} \left(\frac{-(L_{\text{н}} + L_{\text{a},\text{a}}) \sin \alpha \pm X_{R_{\text{min}}} \cos \varphi}{L_{\text{н}}(L_{\text{н}} + 2X_{R_{\text{min}}}) - X_{R_{\text{min}}}^2} \right) \right] \frac{h_{\text{max}}}{K}.$$

Sign (+) in the second bracket is related to the case when ~~capacitor~~ ^{condenser} and adiabatic zone are arranged in the upper half of the steam chamber; sign (-) - to the case when ~~capacitor~~ ^{condenser} and adiabatic zone are arranged below.

For the upper half of the steam chamber

$$X_{R_{\text{min}}} = L_{\text{н}},$$

for the lower half of the steam chamber

$$X_{R_{\text{min}}} = L_{\text{н}} - \frac{gcr' \rho_{\text{ж}}^2 \cos \varphi}{K_{1\mu_{\text{ж}}} q_{\text{ymax}}}. \quad (3.9)$$

If $X_{R_{min}}$ is obtained negative, ^{this} means the minimum radius of curvature of R_{min} is located on interface evaporator ~~capacitor~~ - ~~capacitor~~ ^{condenser} and $X_{R_{min}} = 0$. 2. Adiabatic zone is arranged between the evaporator ~~capacitor~~ ^{condenser} and the ~~capacitor~~ (Fig. 20d), $0 \leq \alpha \leq 90^\circ$:

$$q_{max} = \left[\frac{2\rho_{\kappa}^2 r'}{\mu_{\kappa}} \left(\frac{L_{\kappa}(L_{\kappa} + 2L_{a,3} + 2X_{R_{min}}) - X_{R_{min}}^2}{L_{\kappa}(L_{\kappa} + 2L_{a,3} + 2X_{R_{min}}) - X_{R_{min}}^2} \right) - \right. \\ \left. - \frac{2\rho_{\kappa}^2 r'}{\mu_{\kappa} h_{max}} \left(\frac{-(L_{\kappa} + L_{a,3}) \sin \alpha \pm X_{R_{min}} \cos \alpha}{L_{\kappa}(L_{\kappa} + 2L_{a,3} + 2X_{R_{min}}) - X_{R_{min}}^2} \right) \right] \frac{h_{max}}{K}.$$

Here sign (\pm) in the second bracket has analogous value, as in the preceding case.

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2. Experimental study of cooling ~~the~~ ^{of} semiconductor devices, placed in the capillary-porous cores of ~~the~~ thermal ~~parts~~ ^{tubes} and steam chambers.

This paragraph ~~is~~ is dedicated to the experimental study of

cooling ^{of} ~~the~~ semiconductor devices, placed inside thermal ^{tubes} ~~test~~ and covered with porous core. ~~Were~~ investigated three forms of radiators, having the identical cooling surface: massive radiator, radiator with use of ~~the~~ principle of thermosiphon, and the principle of thermal ~~tube~~ tube,

As fuel element was used ~~the~~ silicon diode D242, having the following operating characteristics: total power 12.5 W, the surface area 8.2 cm², permissible operating temperature to 120°C. Was produced the calculation of radiator as ^a transformer of heat flux ~~at~~ at the given power under normal ambient conditions.

During the calculation the ^{reserve} ~~supply~~ according to the power, scattered by convective heat exchange, ^{we} select ^{as} 200%, i.e., radiator must scatter 25 W. According to formulas for the heat flux, ~~removed~~ removed by convection from the finned surface of radiator [74, 75], it is possible to obtain the effective area of ^{scattering} ~~diffusing~~ surface (see Fig. 19a)

$$F_{\text{эф}} = F_1 + EF_{\text{оп}} = \frac{Q}{\alpha \Delta T} = 350 \text{ cm}^2. \quad (3.11)$$

Here Q is the power, scattered from the surface of radiator convectively; α - the coefficient of ~~the~~ heat exchange of surface with the environment for standard conditions. It is equal to 12 $W/m^2 \cdot ^\circ C$; ΔT - the temperature drop:

$$\Delta T = T_1 - T_2 = 80 - 20 = 60^\circ C.$$

Taking into account the mutual effect of fin^s~~edges~~, their amount is selected in ~~apparitors~~ ^{limits of} 4-5 with the length of fin~~edge~~ 15 mm. The general view of massive radiator is represented in Fig. 19b.

If we assume the absence of the convective heat removal from the external surface of radiator, then according to diagrams [76] it is possible to calculate the power, ~~abstractly~~ removed by emission~~radiation~~.

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during uniform warm-up to $80^\circ C$ radiator can scatter 8 W, while during warm-up to $100^\circ C$ - 10.2 W into unlimited space.

On the basis of that which was stated above, the average density of the heat flux, removed from the surface of radiator by emission/~~radiation~~ and convection, will be equal to

$$q_{\text{cp}} = \frac{Q_{\text{конв}} + Q_{\text{изл}}}{F_{\text{эф}}} = 0,093 \quad \text{W/cm}^2 \quad (3.12)$$

Averaged maximum heat-flux density from the surface of the diode ~~of~~ $q_{\text{д}} = 2,93 \text{ W/cm}^2$. Thus, the ~~common/general~~ total transformation ratio of heat flux for this radiator is equal to

$$n = \frac{q_{\text{д}}}{q_{\text{cp, рад}}} = 31,5. \quad (3.13)$$

The general view of radiators is represented in Fig. 19a, b, c. The third type of radiator, which uses principle of thermal tube, is the radiator, shown in Fig. 19c, internal surface of which and the surface of diode were covered ^{ed} ~~coated~~ with capillary-porous core. The relation of evaporator, ~~vaporizer~~ and ^{condenser} ~~capacitor~~ is equal to 6.5.

Structural calculation of the second and third type of radiators included the selection of working fluid and the calculation of capillary porous core.

The selection of working fluid was conducted from the following considerations: 1). inertness with respect to the materials of equipment, core and chamber walls; 2). the adequate boiling point and the high value of ^{latent} heat of vaporization; 3). low ~~ductility/toughness~~ viscosity; 4). surface tension must be sufficient for the lift of liquid on core ^{during} ~~in~~ chamber operation against the forces of gravitation; 5). thermal resistance.

With heat removal from the open in electrical relation ^{circuits} ~~diagram~~, working fluid must have high dielectric properties.

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The fundamental characteristics of working fluid are Bond's

criterion $Bo = \rho g L^2 / \sigma$ and Koontz' characteristic parameter [6] $N = \rho r' \sigma / \mu$. The criterion Bo is the relation of the forces of gravitation to capillary forces and characterizes the ability of liquid to move over core in the gravitational field. With optimum selection of liquid the value must be minimum. Koontz' parameter N is the complex characteristic of liquid when using it in thermal tube. The optimum selection of liquid is determined by the adequate boiling point and by the maximum value.

The selection of capillary-porous core entails the determination of the maximum ~~altitude~~^{height} of the elevation of liquid from Jurin's formula

$$h_{\max} = \frac{2\sigma \cos \theta}{\rho g r} \quad (3.14)$$

and of the coefficient of permeability

$$K = \frac{j \mu L}{A \rho \Delta P} \quad (3.15)$$

The core, used in thermal tubes, must be dielectric, ~~have~~ *raise* liquid against the forces of gravitation to the height ~~of~~ *of* h_{max} greater than the length of tube. Furthermore, the core must have a minimum of ~~the~~ flow resistance, i e., high K . The ~~better~~ *best* core for application ~~also~~ under these conditions is the package of parallel ~~filaments~~ *fibers*. Usually as core are applied the sintered ~~filaments~~ *fibers*, powders, grids, different fabrics and ceramic ~~etc.~~ *etc.*

~~Concrete~~ specific ~~actual~~ calculation of core for a radiator for the assigned power consists of the following: the mass fluid flow in core is calculated from formula

$$I = \frac{Q}{r} = 1,36 \cdot 10^{-2} \quad g/sec.$$

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For fiberglass materials the porosity is usually equal to 60-70% and the velocity of the motion of liquid at the maximum ~~altitude~~^{height} of absorption ^{is} 16 cm., at height 4 cm^{is} is empirically obtained equal to 0.05 cm/s. Area is calculated from formula

$$S = \frac{j}{V_{0m} \Pi} = 0,5 \text{ cm}^2, \quad (3.16)$$

whence the thickness of core we obtain equal to 0.8 mm.

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The work of thermal ~~heat~~ ^{tube is influenced} ~~affects~~ ^{by} not only the supply of liquid into the zone of evaporation, but also heat supply, since during adverse heat supply in porous core appears the crisis of boiling, which will entail a decrease in the transmitted power and an increase in the temperature of heater, in consequence of which increases ~~by~~ R. Therefore the heater it is expedient to place into core, since this improves heat removal from heat surface.

For the study of this process, was carried out the series of experiments, consisting of two stages. During the first stage were carried out the investigations of the transport properties of core with the maximum heat removal. For this, the heater was placed into core with the previously obtained characteristics. The experiment

showed that the desiccation of core with a thickness 0.8 mm occurred at $Q = 40-50$ W and heat-flux density 3 W/cm^2 . With an increase in the thickness of core to 3 mm, the desiccation of core began with $Q = 200-250$ W. Thus, the calculations obtained experimental confirmation.

The second stage was the experimental study of heat withdrawal during heat supply through the wall. The experiment showed that the crisis of boiling appears at considerably less heat-flux densities. With an increase in the heat flux up to ~~the~~ calculated, strongly increases the temperature differential between the heater and the liquid, which as the final result leads to increase in R . This lowers the effectiveness of the use of a thermal tube.

For a comparative investigation of different types of transformers, was made the experimental installation, which is ~~the~~ ^a chamber, three different radiators and monitoring-measuring equipment. The diagram of experimental set-up is depicted ~~on~~ ⁱⁿ Fig. as 21. Installation made it possible to produce investigations under different ambient conditions. The inspection of the temperatures in the different points of radiators was conducted with the aid of copper-constantan thermocouples. The lay-out diagram of thermocouples is given in Fig. 19b, c. The mean error in the measurement of the temperatures in steady state was 0.2°C . Power input was controlled with maximum error ^{of} 0.01% .
^

The procedure for experiment entailed the following. The temperature characteristics of each of the investigated diodes were ~~measured~~ taken ~~with~~ massive and hollow radiator.

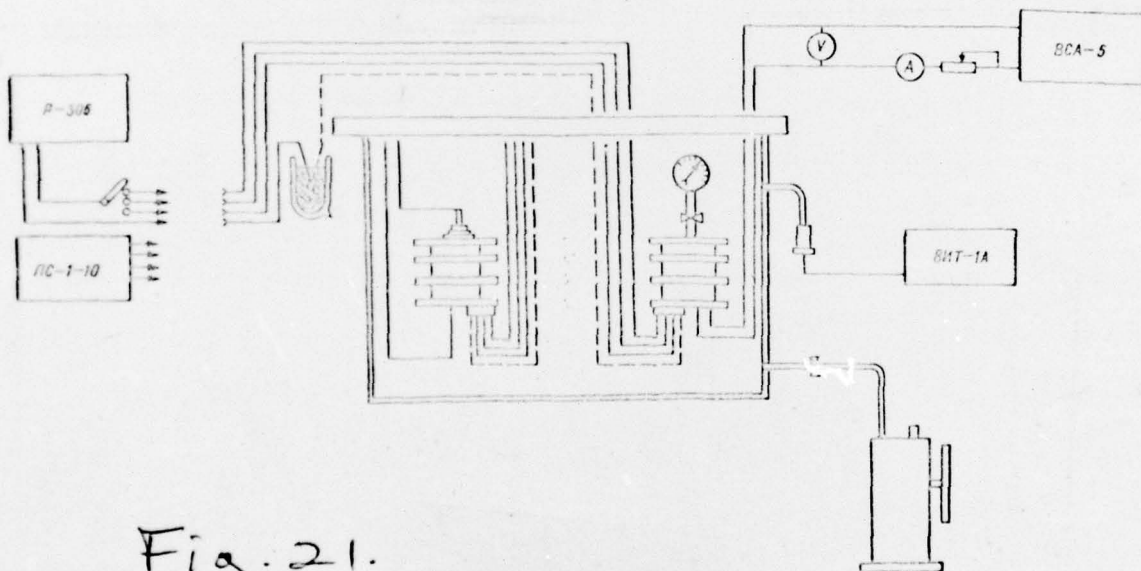


Fig. 21.

Fig. 21. Experimental installation for the investigation of the different forms of radiators.

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After the installation of diode into the small chamber ^{was} filled the necessary amount of water and from the chamber ~~it~~ was forced out air.

Tests on massive and hollow radiators were conducted [#] equally. To heater ~~with~~ ^{by} stages was supplied the power, each following ~~stage~~ ^{was} stage was supplied after ~~will be~~ establish ~~the~~ ^{id} the stationary distribution of temperatures.

Experimental results are given in Fig. 22, 23. ^{Determining} ~~During~~ during experiments was the temperature of the heat-releasing crystal whose value was determined from readings of thermocouple 1 for the diode, ~~established~~ installed on massive radiator and thermocouples 5 for the diode, placed into the hollow chamber.

In Fig. 22 is demonstrated the dependence of the temperature of fuel element from the applied power for the different methods of ~~the~~ heat removal.

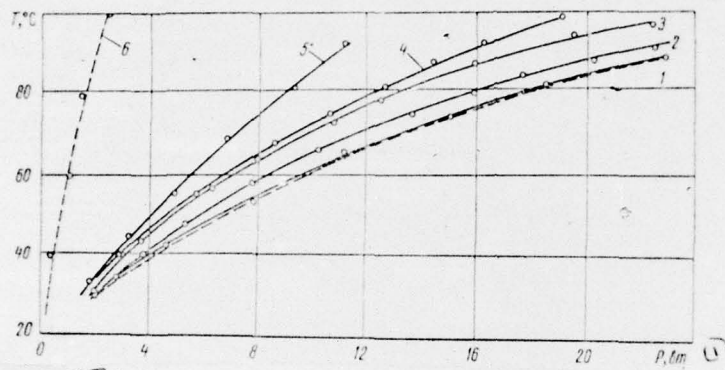


Fig. 22.

Fig. 22. Dependence of the temperature of diode ^{on} the applied power during cooling with the aid of ~~the~~ steam chamber: 1 - in normal

position; 2 - in a horizontal position; 3 - in inverse position; 4 - diode with massive radiator; 5 - diode with hollow radiator in inverted ~~straight~~ position; 6 - diode without radiator.

Key: (1). W.

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The use of water as intermediate heat-transfer agent made it possible by more than 500/o to increase the ~~abstract~~ removed power without an increase in the operating temperature of semiconductor diode, and at power ^{of} 12.5 W its temperature was lowered ^{by} ~~to~~ 10°C, which improves its electrical characteristics and raises the reliability of work. An increase in the effectiveness of heat removal is achieved by a reduction in the resistance of heat transfer from the heat-releasing crystal to the wall of radiator.

Figure 23 gives values R in temperature dependence of heater for massive and hollow radiators.

At ^{low} ~~low~~ temperatures in hollow radiator evaporation ^{is} nonintensive and the heat transfer is ~~realized~~ accomplished in essence by thermal conductivity of liquid; therefore ⁱⁿ the initial section the massive radiator has effectiveness even somewhat above.

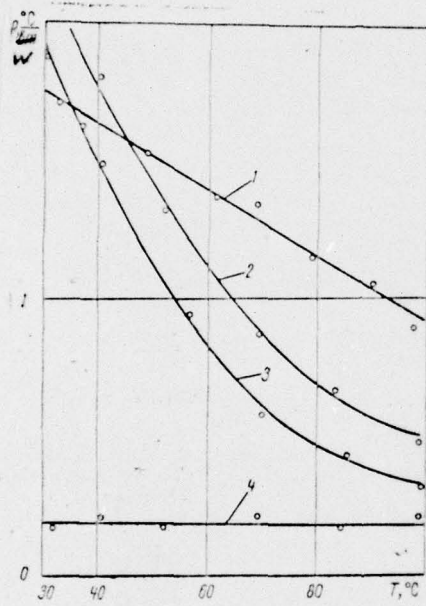


Fig. 23.

Fig. 23. Value in temperature dependence of heater for the massive and hollow radiators: 1 - $R = T_1 - T_3 / Q$; 2 - $R = T_5 - T_7 / Q$; 3 - $R = T_2 - T_3 / Q$; 4 - $R = T_3 - T_7 / Q$.

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Then with an increase in the temperature of heater ^{the} liquid begins ever more intense ^{ly} to be vaporized and the effectiveness of evaporative system is raised. Fig. 23 gives changes in thermal resistance $R_1 = T_5 - T_7 / Q$, $R_2 = T_2 - T_3 / Q$ and $R_3 = T_3 - T_7 / Q$. From examination it may be concluded that at these powers a system of the type of thermal tube works still not ⁱⁿ the most effective form, since, obviously, optimum will be this position in which $R_2 - R_3$ is minimal. This will determine the optimum working temperature of device. Near this point evaporative system will work in the ~~more~~ conditions of thermal tube and thermal resistance between the ^{area} ~~range~~ of heating and the ^{area} ~~range~~ of condensation will be negligible.

Introduction to the chamber of capillary-porous core makes it possible to orient it in any direction. Figure 22 gives the dependence of the temperature of diode ^{on} ~~from~~ the applied power for three different orientations. Even in position with heater ~~the~~ above ^{the} hollow chamber with core ~~abstract~~ removes heat from the heated

~~core~~ element better than massive radiator. A sufficiently powerful change of the characteristics depending on orientation can be explained by the loose fit of core to the surface of diode. But also in this position the effectiveness of heat withdrawal when using a principle of thermal tube is higher than ^{for a} ~~of~~ massive radiator (Fig. 22).

The location of fuel element inside tube makes it possible to avoid the appearing in fine vacuum influence of contact resistance. The experiments, carried out in vacuum, ~~they~~ showed the influence of ~~the~~ contact resistance, appearing between the diode and the radiator. The effectiveness of the work of radiator decreased 1.5 times, and hollow radiator with liquid - 1.2 times.

The experiments conducted prove the advantages of heat withdrawal from the heated ~~core~~ elements, placed inside the porous core of tube. By utilizing this principle of cooling semiconductors, it is possible to considerably lower the operating temperature of ~~core~~ element and thereby to raise the stability of its work. During practical application ~~the~~ important value acquires the weight of equipment. On a contemporary level of development ^{of} radio electronics, with increase in the ~~abstract~~ removed heat fluxes the cooling radiators can have considerably larger ¹ weight, than most cooled construction. Hollow radiator, utilized in experiments, ^{1.5} 1.5 times ₁

lighter than the solid one.

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Thus, the location of the heat-releasing systems into thermal tubes with dielectric liquid, besides an improvement in the temperature characteristics, will make it possible to considerably lower total construction weight, and capillary-porous core allow ~~assumes~~^S freedom in the orientation of the cooled system.

3. Cryogenic thermal ~~tubes~~^{tubes}

The experimental study of the parameters of cryogenic thermal ~~duct~~^{tube} with the use of liquid nitrogen as heat-transfer agent is described in work [58, 98].

The maximum heat flux, transferred along cryogenic thermal ~~duct~~^{tube}, according to Kutter's theory [1], with some assumptions [33] it is possible to find from formula

$$Q_{\max} = N_{\text{м}} \delta \left(\frac{8\pi r_{\text{вн}}^2 r_{\text{м}}^2}{b r_{\text{эф}} (L_{\text{н}} + 2L_{\text{а}} + L_{\text{м}})} \right), \quad (3.17)$$

$$N_{\text{м}} = \frac{\rho_{\text{м}} N_{\text{м}}' r'}{\mu_{\text{м}}},$$

where b is a coefficient of permeability; $r_{\text{вн}}$ - the inside radius of wall; $r_{\text{эф}}$ ~~are an~~ effective radius of pores; $r_{\text{м}}$ - ~~an~~ effective capillary radius for a fluid flow; δ ~~is~~ thickness of porous core; $N_{\text{м}}$ - the criterion, which characterizes the transport properties of liquid.

The gradient of temperature is determined from known heat transfer rate and the thermal resistance of the core, filled by the liquid:

$$\frac{\partial T}{\partial y} = \frac{q}{\lambda_{\text{эф}}}. \quad (3.18)$$

The effective thermal conductivity of the porous core, filled by liquid, can be ^{calculated} ~~designed~~ according to formula from work [59]

$$\frac{\lambda_{\text{eff}}}{\lambda_{\text{en}}} = \frac{1}{\frac{1}{(h/L)^2} + A} + v_r \left(1 + \frac{h}{L} \right)^2 + \frac{2}{1 + \frac{h}{L} + \frac{1}{v_r h/L}} \quad (3.19)$$

~~Page 95.~~

Here

$$A = \frac{1}{\frac{\lambda_m}{\lambda_{ch}} + \frac{v_{r,3}\pi}{16k_m k_m} \left(\frac{h}{L}\right)^2 10^3},$$

$$L = l + h, \quad h/L = \frac{h/l}{1 + h/l}, \quad v_r = \frac{\lambda_r}{\lambda_{ch}}, \quad v_{r,3} = \frac{\lambda_{r,3}}{\lambda_{ch}},$$

$$k_m = \frac{h_m}{L} \cdot 10^3,$$

l is the basic dimension of the pore of porous material; L - the overall size of the unit cell of porous system; $h = 2\Delta$ - the thickness and the width of the rod of the solid skeleton of cell; k_m - the coefficient ~~is~~, which characterizes cohesion ~~surplus~~ ^{of} the microroughnesses of two adjacent particles; $\lambda_{r,3}$ - the thermal

conductivity of gas microgap; λ_{ch} - the thermal conductivity of the skeleton of porous material; λ_{tc} ~~and~~ contact thermal conductivity; h_m - the height ~~and~~ of microroughnesses; λ_r - the coefficient of the thermal conductivity of gas.

For the thermal ~~not~~ ^{tube} of ~~constant~~ invariable geometric parameters, the use of different liquids will lead to the fact that the maximum heat flux ~~varies~~ in proportion to a change in N_{re} and $1/\lambda_{\text{gph}}$ of core.

For the assigned heat flux ~~of~~ Q_{max} , the use of different liquids will lead to the fact that will have to change the dimensional characteristics of core, in particular, its thickness δ inversely proportional ~~to~~ N_{re} ^{to} and the use ^{of} different liquid ~~will~~ ^{will} lead to change ^{of} dT/dy due to a change in ~~the~~ λ_{gph} . This can be reflected by the term ~~of~~ $\lambda_{\text{gph}} N_{\text{re}}$.

From the aforesaid it follows that the cryogenic thermal ~~not~~ ^{tubes are} not in state to transfer large heat fluxes, furthermore, in them occur ~~the~~ high gradients of temperature.

Thus, for instance, in work [58] the temperature differential along nitrogen thermal ~~not~~ ^{tube} was 20° during the transmission of heat output ^{of} 20 W. In work [2] the temperature differential on freon (Freon ^{of} Λ)

22, Freon-11) thermal ~~ducts~~^{tubes} was 30° during the transmission of heat output ^{of} 15 W. Consequently, for cryogenic thermal ~~ducts~~^{tubes} is not applicable the concept of isothermal device, as this is assumed to be in Kutter's theory [6, 8, 9].

The operating temperature of cryogenic thermal ~~ducts~~^{tubes} can be located within temperature range between the temperature of triple point and critical temperature. For cryogenic liquids this temperature range is sufficiently narrow.

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Thus, for instance, for liquid nitrogen it is 63°K . Consequently, the dependence of the properties of liquid on temperature must be considered during the analysis of the transport properties of thermal ~~ducts~~^{tubes}. The values of the coefficient of surface tension σ and of thermal conductivity λ ^{of} liquid nitrogen change in the range of temperatures from 70 to 80°K , from 10, 53 to 8, 27 dyn/cm and from $1.52 \cdot 10^4$ to $1.36 \cdot 10^4$ erg/cm. At the same time a change in the properties of vapor phase ~~from~~^{from} temperature does not have vital importance.

In cryogenic thermal ~~ducts~~^{tubes} the pressure differential in vapor phase ^{is} always less than the pressure differential, which appears as a

result of the action of capillary forces; however, for ~~the~~ simplicity of their analysis can be considered equal to each other

$$\Delta P_{\pi} = \Delta P_{\text{nan}} = \frac{2\sigma}{R_{\text{min}}} \quad (3.20)$$

Correspondingly

~~respectively~~ the temperature differential in vapor phase, according to the equation of ^{Clausius}~~Klauiusa~~ - Clapeyron,

$$\Delta T_{\pi} = \frac{RT_{\pi}^2}{P_{\pi} r'} \cdot \frac{2\sigma}{R_{\text{min}}} \quad (3.21)$$

The value of ΔT_{π} is usually low, since ~~the~~ T_{π} and σ are small, but P and r' are great. Thus, for instance, for the saturated nitrogen at ~~the~~ atmospheric pressure ~~of~~ ($R_{\text{min}} = 50 \mu$) $\Delta T_{\pi} \approx 0,03^{\circ}\text{K}$.

For the determination of the special feature ~~realities~~ of

the distribution of temperature field in the core of the cryogenic thermal tube, which has liquid cooler in the zone of ~~capacitor~~^{condenser}, was made ~~the~~^{an} experimental installation (Fig. 24). As heat-transfer agent of thermal tubes were selected ~~the~~ Freon-22 and ~~the~~ Freon-11, possessing low thermal conductivity and high viscosity.

Thermal ~~unit~~^{tube} had the following parameters: length ~~are~~ 1.8 m; outside diameter - 19.5 mm; the wall thickness of tube - 0.25 mm (stainless steel). In ~~unit~~^{tube} was located porous core from glass cloth 3.5 mm in thickness. Its characteristic: $R_{min}=4 \cdot 10^{-5} \text{ M}$ and $K_1=0.25 \cdot 10^{11} \text{ M}^2$. The ~~inspection~~^{control} of temperatures is conducted by the differential copper-constantan thermocouples, stuck on the surface of tube, and by the copper thermometer-resistance^s, located: on the surface of ~~unit~~^{tube} and within core in the zones of evaporation and condensation.

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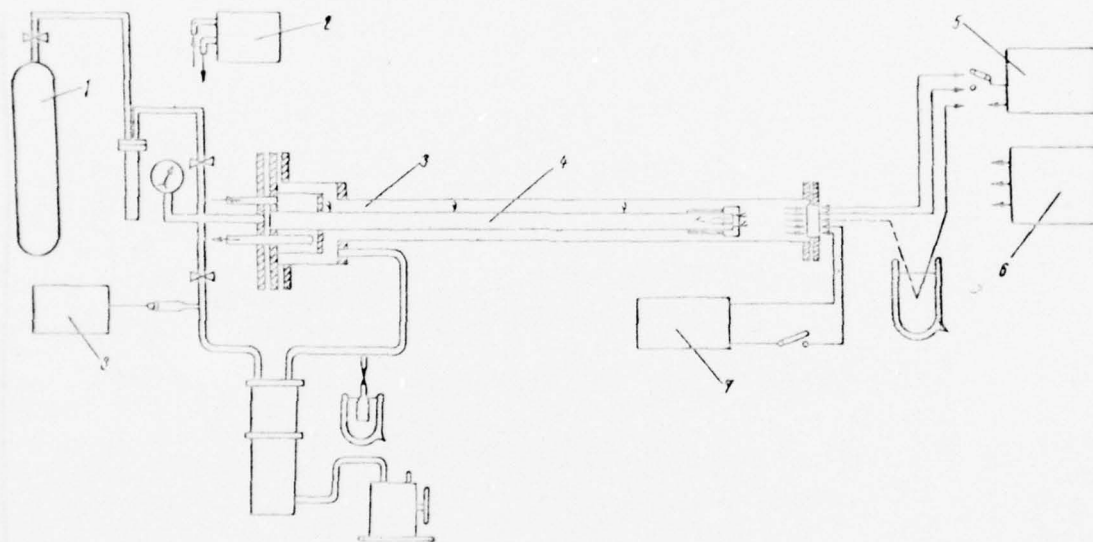


Fig. 24.

Fig. 24. Diagram of experimental installation for the study of the low-temperature thermal ^{tube} ~~data~~; 1 - tank with liquefied gas; 2 - thermostat; 3 - vacuum chamber; 4 - the investigated ^{tube} ~~data~~; 5 - low-resistance potentiometer; 6 - the recording potentiometer; 7 - the source of power; 8 - vacuum gauge.

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Furthermore, the temperature ^{of vapor} ~~pair~~ in the zone of evaporation is measured by thermistor. For the elimination of the convective heat exchange between the surface of tube and the environment, the tube was placed into vacuum seal. In vacuum shell is supported the ~~evacuation~~ rarefaction ^{of} 10^{-4} mm.

Were carried out the calculations of the heat-transmitting ability of ^{tube} ~~data~~ according to Kutter's formula [1] with its filling with different liquids. It turned out that the greatest heat-transmitting ability possess water and ammonia, and the smallest heat fluxes it is possible to transfer when using Freon^s. But ammonia has a series of deficiencies ^{its} ~~data~~, which impede its use. It is poisonous, requires special instrumentation and, furthermore, it reacts with some metals. Water cannot be used at minus temperatures. For experimental study as heat-transfer agents, were selected ~~data~~

Freon-11 and Freon-22. Calculations according to formula (3.17) gave the values of the maximum transferred heat output: for Freon-11 $Q_{\max} = 1.75$ W at 30°C and for Freon-22 $Q_{\max} = 1.26$ W with 30°C . The calculations are ~~produced in connection with~~ ^{made relative to} the temperature of the heat-insulated zone. It is necessary to note that the maximum transmitted power increases with lowering in the operating temperature of tube. Heat-input with emission ~~radiation~~ to the surface of tube depending on operating temperature varies from 4 to 1.5 W.

Figure 25 gives the distributions of temperatures along the length of tube for the different transmitted powers and slope angles. As working liquid serves Freon-22. In ~~capacitor~~ ^{condenser} was supported temperature -25°C . The temperature differential near ~~capacitor~~ ^{condenser} almost completely determines entire temperature drop along transport zone.

Fig. 26 represents the experimental dependences of the operating temperature of the tube, filled by Freon-22 and Freon-11, ~~from~~ ^{on} the applied power at different slope angles. In a horizontal position the maximum power, transferred by tube, is equal to 15, i.e., almost 2 times ~~is~~ more than it follows from Kutter's formula. Temperature ^{of} the ~~liquid~~ ^{vapor} and the temperature, measured in a series of points on the surface of tube, are close to each other in the case of the positive angles of ~~the~~ slope of tube, which indicates a good thermal insulation of adiabatic zone.

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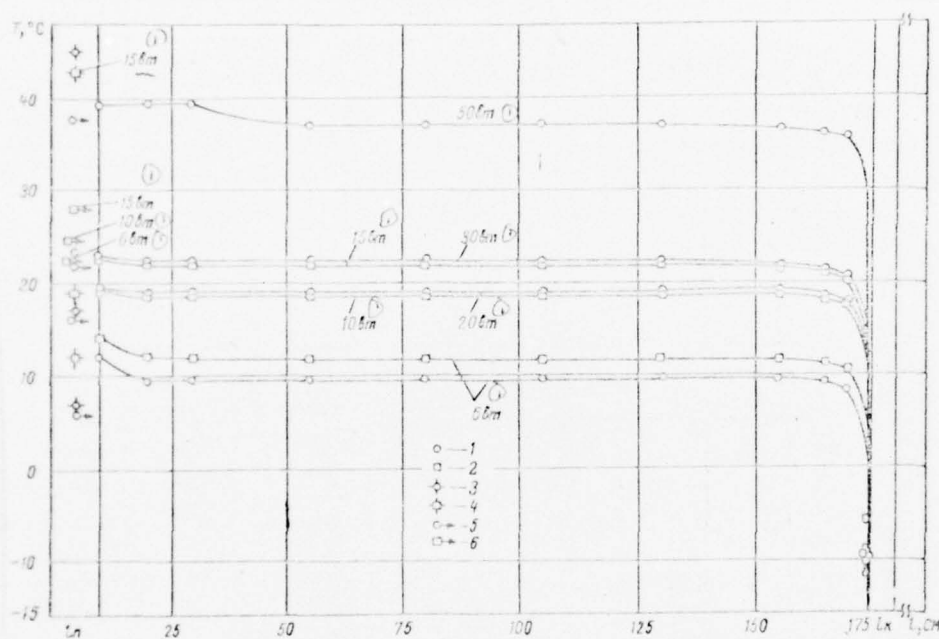


Fig. 25.

Fig. 25. The distribution of temperature along the length of the tube, which uses Freon-22; 1 - $\frac{\alpha}{A} = +5^\circ$; 2 - $\frac{\alpha}{A} = 0$; 3, 4 - the temperature of ~~of~~ liquid in core in the zone of evaporation; 5, 6 - temperature ^{of vapor} ~~in the~~ in the zone of evaporation.

Key: (1). W.

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^{with} ~~is~~ the work of tube in a horizontal position ($\phi = 0$) the vapors of working fluid are considerably overheated relative to the temperature of the wall of tube. This overheating reaches 10° .

Figure 27 gives the dependences of the working temperature of tube and the temperature differential along transport zone as function of the temperature of liquid in the ^{condenser} ~~capacitor~~ of the tube, which works at ^a ~~the~~ positive angle of ~~the~~ slope with ~~of~~ the transferred on tube heat output ^{of} 20 W.

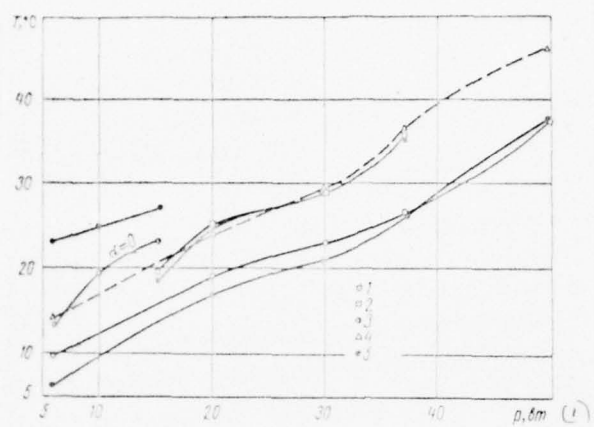


Fig. 26.

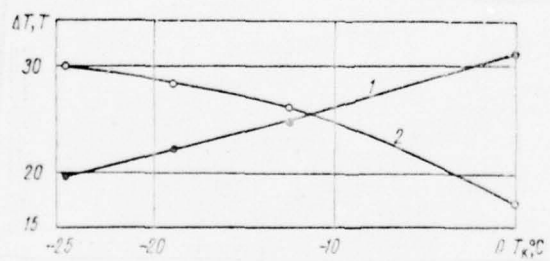


Fig. 27.

Fig. 26. Dependence of operating temperature ~~from~~^{on} the power: 1 - $\alpha = +5^\circ$, working fluid Freon-22; 2 - $\alpha = +2^\circ$, Freon-22; 3 - $\alpha = 0$, Freon-22; 4 - $\alpha = 0$, Freon-11; 5 - the temperature ~~of~~^{of vapor} in the zone of evaporation.

Key: (1). W.

Fig. 27. Dependence of operating temperature (1) and of temperature differential along transport zone (2) ~~from~~^{on} the temperature of liquid in ~~capacitor~~^{condenser}.

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By an increase in the temperature of ~~capacitor~~^{condenser} it was possible to raise the effective thermal conductivity of tube almost 2 times, in this case considerably increased the temperature of the surface of tube. A further increase in the temperature of ~~capacitor~~^{condenser} turned out to be impossible due to too high an operating pressure.

Freon-11 in its thermophysical characteristics is close to Freon-22. By ~~this~~^{this} it is possible to explain the fact that the dependences of temperature ~~from~~^{on} transmitted power for the tube, which uses as

heat-transfer agent Freon-11 and Freon-22, take the same form (see Fig. 25). Curves are obtained at identical positive slope angle and the identical temperatures of ~~capacitor~~ ^{condenser}. The operating temperatures of tube ~~are distinguished~~ ^{differ} by approximately 10°.

It is necessary, however, to consider that during the replacement of Freon-22 by Freon-11 the operating pressure in tube descends ~~from~~ ^{from} 15 atm. to 1 atm., which raises reliability and the safety of the work of thermal tube.

On the basis of the experiments conducted it is possible to make the following conclusion. When using in thermal ducts as the heat-transfer agent ~~of the~~ liquids, which possess ~~small~~ ^{low} thermal conductivity, in the ~~capacitors~~ ^{condensers} of thermal ~~tubes~~ ^{tubes} occurs an essential temperature drop. This is correct in the presence of ~~the~~ high coefficient of heat exchange on the external surface of thermal ~~tube~~ ^{tube} in the zone of condensation.

With a temperature decrease in the heat exchanger of ~~capacitor~~ ^{condenser}, increases the temperature drop in ~~capacitor~~ ^{condenser, and}, deteriorate the working conditions of thermal ~~tube~~ ^{tube}. The published in the literature ~~calculations~~ ^{calculations} equations for determining the maximum heat output, transferred on low-temperature thermal ~~tube~~ ^{tube}, considerably differ from the experimentally determined values.

The thermal tubes, which use as heat-transfer agents Freon, can be recommended for cooling everyday coolers, and also radio-electronic equipment with comparatively low heat-flux densities. Freon⁹ are chemically inert and do not interact with the materials of tube and equipment. From the comparison of the work of the tube, which uses Freon-22 and Freon-11, it is possible to ~~give the~~ ^{give the} advantage to Freon-11 from the considerations of the safety of work. By a change in the characteristics of core and by a contraction in length of tube it is possible to considerably increase the transferred heat fluxes.

Large prospects have cryogenic thermal ~~units~~ ^{tubes} when using them as thermal key ~~branches~~ ^S in cryogenic electric power lines and ~~the~~ superconducting electrical machines and solenoids. In this case can effectively be utilized both the thermal ~~units~~ ^{tubes} and thermosiphons [99, 98].

Interesting possibilities ~~have~~ ^{exist} for using cryogenic thermal ~~units~~ ^{tubes} in medicine and biology [101, 103].

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~~Page 102.~~4. Coaxial thermal ^{tubes.} ~~tubes.~~

In a number of cases, thermal ^{tubes} ~~units~~ can effectively be used as thermal transformers for the ~~purpose~~ purpose of concentration or deconcentration of heat flux. ^{This} ~~the~~ the concentration of ~~the~~ heat flux, brought to the earth ~~by~~ by solar radiation, makes it possible to create effective electric power sources (thermoelectric batteries), to heat water in heat exchangers. The concentration of the thermal radiation of Sun or other sources of infrared (thermal) radiation can be realized ~~accomplished~~ both with the aid of ~~the~~ optical means and with the aid of thermal transformers of the type of ~~the~~ thermal ^{tubes} ~~units~~.

and steam chambers. Thermal transformers ~~expensive~~ⁱⁿ the form of ~~the~~ thermal ~~units~~^{tubes} and steam chambers can be utilized not only for purposes of the concentration of the thermal radiation of the Sun, but also for the concentration of ~~the~~ heat flux, ~~isolated~~^{given off}, for example, by isotopic radioactive ~~active~~ elements, etc.

~~The~~ Even greater application ~~use~~ thermal transformers can find as the means for the deconcentration of heat flux, which is very important for cooling a series of ~~the~~ heat-releasing objects.

The deconcentrators of heat flux, or peculiar emitters, were widely applied, for example in space technology where the heat withdrawal into the environment is realized ~~accomplished~~ in essence by ~~emission~~ radiation. The thermal ~~units~~^{tubes} and the steam chambers, made as the deconcentrators of heat flux, can successfully be used for cooling the radioactive fuel elements in atomic reactors, ~~the~~ optical heat-releasing devices, electronic devices, etc.

Figure 28 shows the different constructions of thermal transformers.

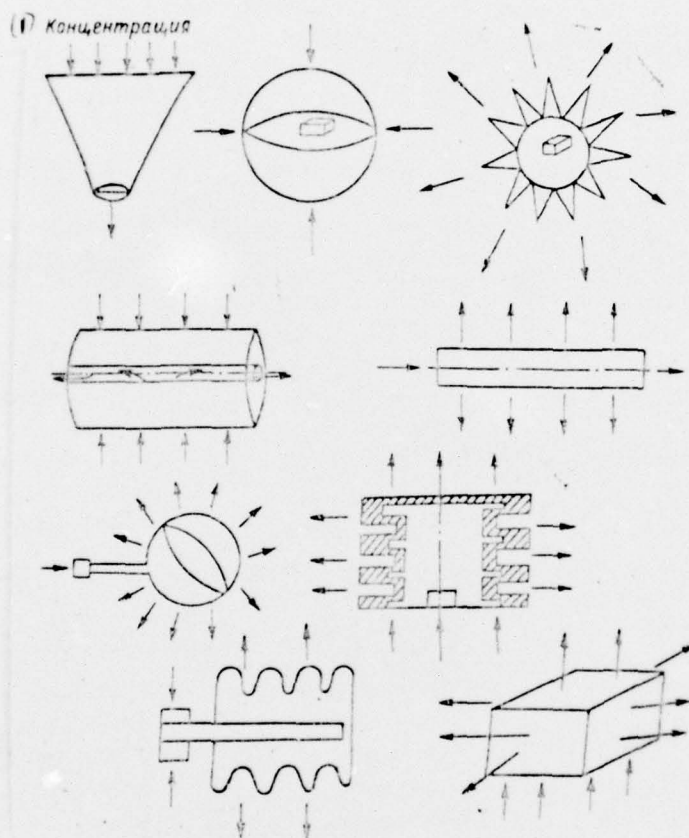
~~Page 103.~~

For cooling and heating of the cylindrical surfaces, which have

the high value of the ratio of length to the transverse
~~size~~ dimension L/d (^{tubes}~~ducts~~ and rods), with the aid of a thermal
transformer of the type of thermal ^{tube}~~duct~~, the most successful
construction is coaxial thermal ^{tube}~~duct~~ [85, 104].

Fig. 28. Different constructions of thermal transformers.

KEY: (1). Concentration.



~~Pages 104 and 105.~~

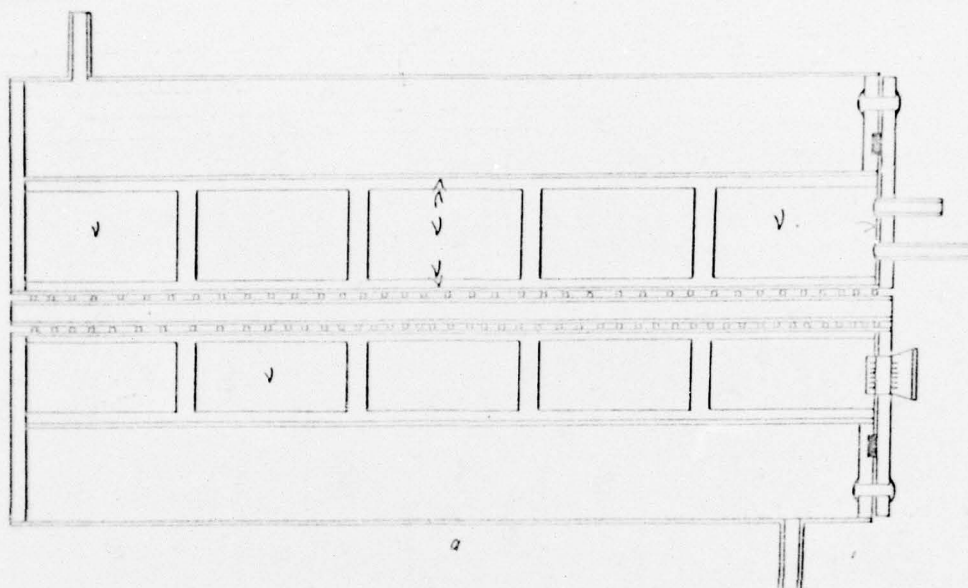


Fig. 29.

Fig. 29. one of the versions of the construction of coaxial thermal ~~duct~~ ^{tube} (a) and the ~~cell~~ ^{tube} element of coaxial thermal ~~duct~~ (b): 1- core; 2 - the film of liquid; 3 - adiabatic zone; 4 - core in the zone of evaporation; 5 - ~~steam~~ ^{vapor} space; 6 - the wall of ~~capacitor~~ ^{condenser}; 7 - the wall of evaporator ~~vaporizer~~.

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Figure 29 shows one of the versions of the construction of coaxial thermal ~~duct~~ ^{tube}. The evaporator ~~vaporizer~~ and the ~~capacitor~~ ^{condenser} of this type of thermal ~~duct~~ ^{tube} have the cylindrical surface of approximately equal length, but different diameter and are inserted one in another (~~duct~~ ^{tube} in ~~duct~~ ^{tube}). Transfer ~~heat~~ ^{of vapor} and liquids is ~~realized~~ accomplished in radial direction. For this purpose between the evaporator ~~vaporizer~~ and the ~~capacitor~~ ^{condenser}, are arranged ~~the~~ cavities in the form of ~~the~~ toroids, walls of which are the porous toroidal bushings - capillary pumps. Besides ~~these~~ ^{these} porous bushings, the walls of evaporator ~~vaporizer~~ and ~~capacitor~~ ^{condenser} are covered with ~~thin~~ thin porous core in the form of grid, grooves in the wall of the housing of thermal ~~duct~~ ^{tube} etc.

Since such thermal ~~ducts~~ ^{tubes} can be utilized both under conditions of weightlessness and in gravitational field, porous toroidal

bushings it is possible to connect with each other by ~~the~~ porous fin~~edges~~^s, arranged ~~located~~^{located} by azimuth parallel ~~to~~^{to} flow lines. These fin~~edges~~^s will ensure the lift of liquid against gravitational forces.

In the coaxial thermal ~~unit~~^{tube} or the steam chamber it is possible to distinguish three zones: 1). cylindrical ~~capacitor~~^{condenser} with the geometric dimensions of r_0^k, r_i^k and L_n , 2). adiabatic zone with the geometric dimensions of r_0^n, r_i^n, L_a ; 3). cylindrical evaporator ~~evaporizer~~ with the ~~size~~ dimensions of r_0^n, r_i^n, L_n .

Let us make the following assumptions.

1. The core is incompressible and on the sections of L_n, L_a, L_n has the constant thickness c . Capillary-porous body is isotropic, in any section the area of the pores of the core S_n and the total area S are ~~located~~ found in the relationship

$$\frac{S_n}{S} = \Pi.$$

Analogous relationship ~~is~~ retained for bulk porosity.

2. ^{Vapor} ~~Pairs~~ and liquid through entire length of the ^{condenser} ~~capillary~~ of L_R , evaporator ~~vaporizer~~ of L_n and adiabatic zone of L_a is found at constant temperature, there is no supercooling and overheating of liquid, the vapor pressure $P_n = \text{const}$.

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3. ^{Vapor} ~~Pairs~~ is condensed on interface liquid - ^{vapor} ~~pairs~~ and has a rate of U_n in the direction, normal to the surface, i.e., the given rate of U_n does not have components along the axis Z, respectively there is no change in the momentum along the axis Z.

4. The rate of the liquid, which flows in porous core is equal to U_n and has only Z component, it is constant on the entire thickness and ^{to} ~~the~~ equal average speed of motion ^{of} ~~the~~ liquid in one capillary.

5. The effect of gravitational field we disregard.

Since porous toroidal bushings (adiabatic zones) ~~they~~ divide ^{the} coaxial ^{tube} ~~into~~ into a series of independent sections, ^{it is} ~~sufficient~~ sufficient to examine ~~of~~ one section.

6. All terms, which contain the differentials of the second and ~~of~~ higher orders, also we disregard.

7. On the surface of core, there is no fluid film, condensation ~~is realized~~ ^{of vapor} ~~is realized~~ ^{is accomplished} directly in pores, and evaporation - from the pores of core.

We will examine consecutively three ~~cells~~ elements of core.

Condenser.
~~Capacitor~~. Figure 29b depicts the ~~cell~~ element of coaxial ~~capacitor~~ ^{tube}.
Let us examine the balance of mass and energy for the ~~cell~~ element of porous ~~capacitor~~ ^{condenser} with a length of dz , the external diameter of $2r_0^k$ and inner diameter of $2r_i^k$.

Balance of mass

$$j_{m(1)} + j_n = j_{m(2)}, \quad (3.22)$$

$$j_{m(1)} = \rho_m \pi (r_i^{k2} - r_0^{k2}) U_m \Pi, \quad (3.23)$$

$$U_{m(2)} = U_{m(1)} + dU_m, \quad (3.24)$$

$$j_{m(2)} = \rho_m \pi \Pi (r_i^{k2} - r_0^{k2}) U_{m(2)} = \rho_m \pi \Pi (r_i^{k2} - r_0^{k2}) \times \\ \times (U_{m(1)} + dU_m), \quad (3.25)$$

$$j_n = \rho_n U_n \Pi 2\pi r_0^k dz = j_{m(2)} - j_{m(1)} = \\ = \rho_m \pi \Pi (r_i^{k2} - r_0^{k2}) \frac{dU_m}{dz} dz, \quad (3.26)$$

$$F_{P_1} - F_{P_2} - F_{\tau D} = \rho_m U_1^2 \Pi \pi (r_i^{k2} - r_0^{k2}) - \\ - \rho_m U_2^2 \Pi \pi (r_i^{k2} - r_0^{k2}) = \rho \frac{d(U_m^2)}{dz} dz \Pi \pi (r_i^{k2} - r_0^{k2}). \quad (3.27)$$

~~Page 108.~~

In accordance with the law of Darcys

$$F_{\tau p} = K_1 \frac{\mu_{\text{ж}}}{\rho_{\text{ж}}} j_{\text{ж}} \Pi dz = K_1 \Pi^2 \pi (r_l^{\kappa^2} - r_0^{\kappa^2}) \mu_{\text{ж}} U_{\text{ж}} dz, (3.28)$$

$$F_{P_1} = \left(P_n - \frac{2\sigma}{R} \right) \Pi \pi (r_l^{\kappa^2} - r_0^{\kappa^2}), \quad (3.29)$$

$$F_{P_2} = \left(P_n - \frac{2\sigma}{R + dR} \right) \Pi \pi (r_l^{\kappa^2} - r_0^{\kappa^2}). \quad (3.30)$$

Substituting (3.28) - (3.30) in (3.27)), we obtain

$$-2\sigma \frac{dR}{R^2} - \Pi K_{1m} U_m dz = \rho_m \frac{d(U_m^2)}{dz} dz. \quad (3.31)$$

Energy balance in the ~~cell~~ ^{condenser} element of the porous ~~capacitor~~ of coaxial thermal ~~core~~ ^{tube} we will examine when thermal energy in porous core is transferred by convection; therefore ~~by~~ the heat transfer ^{by} thermal conductivity we disregard:

$$Q_{m(1)} + Q_n = Q_{m(2)} + Q_1, \quad (3.32)$$

$$Q_{m(1)} = j_m h_m = h_m \rho_m \Pi \pi (r_l^2 - r_0^2) U_{m(1)}, \quad (3.33)$$

$$Q_n = j_n h_n = h_n \rho_n \Pi \pi (r_l^2 - r_0^2) \frac{dU_m}{dz} dz, \quad (3.34)$$

$$Q_{m(2)} = j_m h_m + \frac{d(j_m h_m)}{dz} dz = h_m \rho_m \pi \Pi (r_l^2 - r_0^2) \times$$

$$\times \left(U_m + \frac{dU_m}{dz} dz \right), \quad (3.35)$$

$$Q_1 = q 2\pi r_l^2 dz. \quad (3.36)$$

Having

summed up (3.33) - (3.35), (3.36), we will obtain

$$\begin{aligned}
 & (h_n - h_m) \rho_m \Pi \pi (r_l^{K^2} - r_0^{K^2}) \frac{dU_m}{dz} = \\
 & = \rho_m \Pi \pi (r_l^{K^2} - r_0^{K^2}) U_m \frac{dh_m}{dz} + q2\pi r_l^K, \quad (3.37)
 \end{aligned}$$

$$\begin{aligned}
 & U_m \frac{dh_m}{dz} \rho_m \Pi \pi (r_l^{K^2} - r_0^{K^2}) - (h_n - h_m) \rho_m \times \\
 & \times \Pi \pi (r_l^{K^2} - r_0^{K^2}) \frac{dh_m}{dz} + q2\pi r_l^K = 0. \quad (3.38)
 \end{aligned}$$

Since $\frac{dh_m}{dz} = 0$

$$r' \rho_m \Pi \pi (r_i^{\kappa^2} - r_0^{\kappa^2}) \frac{dU_m}{dz} = q 2 \pi r_i^{\kappa}, \quad (3.39)$$

$$\frac{dU_m}{dz} = \frac{2 q \pi r_i^{\kappa}}{r' \rho_m \Pi \pi (r_i^{\kappa^2} - r_0^{\kappa^2})}. \quad (3.40)$$

After integrating expression (3.40) from 0 to z , we obtain

$$U_m = \frac{2 q r_i^{\kappa}}{r' \rho \Pi (r_i^{\kappa^2} - r_0^{\kappa^2})} z, \quad (3.41)$$

$$j_m = \rho_m \Pi \pi (r_i^{\kappa^2} - r_0^{\kappa^2}) U_m = \frac{2 q \pi r_i^{\kappa}}{r'} z. \quad (3.42)$$

The ~~special~~/general/~~total~~ integral equation of energy transfer of substance and momentum in the ^{condenser}~~capacitor~~ of coaxial thermal ^{tube}~~duct~~ takes the form

$$\begin{aligned}
 & - \int_{R=0}^{R=L_n} 2\sigma \frac{dR}{R^2} - \int_0^{L_n} K_{1,1111} \frac{2qr_i^{\kappa}}{r_i^{\kappa} \rho_{11} (r_i^{\kappa^2} - r_0^{\kappa^2})} dz = \\
 & = \int_0^{L_n} \frac{4q^2 r_i^{\kappa^2}}{r_i^{\kappa^2} \rho_{11} \Pi^2 (r_i^{\kappa^2} - r_0^{\kappa^2})^2} dz. \quad (3.43)
 \end{aligned}$$

Adiabatic zone. In adiabatic zone the transfer of liquid is realized ~~by the action of pressure gradient~~ under the action of pressure gradient and can be described by the law of Darcy.

$$j_m = - \frac{2\pi\delta \left(\frac{2\sigma}{R_i^2} - 2\sigma/R_i \right) \rho_m}{K^2 \ln(r_o^2/r_i^2) \mu_m} \quad (3.44)$$

~~Page 220.~~ Since the fluid flow at output ~~from the capacitor~~ ^{condenser} of thermal ~~tube~~ ^{tube} is equal to fluid flow through the adiabatic zone, ~~as then~~ ^{from} this equality it is possible to determine the pressure differential along the length of adiabatic zone, necessary for the transfer through it ^{of} this flow:

$$\frac{2q_{\kappa} r_i^{\kappa}}{r^2} L_{\kappa} = - \frac{2\pi\delta \left(\frac{2\sigma}{R_i^{\kappa}} - \frac{2\sigma}{R_i^{\kappa}} \right) \rho_{\kappa}}{K_2 \ln(r_0^{\kappa}/r_0^{\kappa}) \mu_{\kappa}}, \quad (3.45)$$

$$\Delta P_{a,2} = \left(\frac{2\sigma}{R_i^{\kappa}} - \frac{2\sigma}{R_i^{\kappa}} \right) = - \frac{q_{\kappa} r_i^{\kappa} L_{\kappa} K_2 \ln(r_0^{\kappa}/r_0^{\kappa}) \mu_{\kappa}}{r^2 \delta \rho_{\kappa}}, \quad (3.46)$$

The evaporator ~~evaporizer~~ of coaxial thermal ^{tube.} porous core in evaporator ~~evaporizer~~ has ~~size~~ dimensions of $r_i^{\kappa}, r_0^{\kappa}, L_{\kappa}$. At output ~~field~~ into the evaporator ~~evaporizer~~ with ~~of~~ $Z = L_{\kappa}$, the speed of the motion of liquid is equal to

$$U_{\kappa(z=L_{\kappa})} = \frac{j_{\kappa}}{\rho_{\kappa} S_{\kappa}} = \frac{2q_{\kappa} r_i^{\kappa} L_{\kappa}}{r^2 \rho_{\kappa} \Pi(r_i^{\kappa^2} - r_0^{\kappa^2})}. \quad (3.47)$$

Since it is proposed that the heat flux will be fed to evaporator ~~surface~~ evenly ^{over} ~~the~~ entire area, and the evaporation occurs from the surface of core, it is possible to consider that in evaporator ~~surface~~ occurs the following proportionality:

$$\frac{U_{\kappa(z=L_n)}}{U_{\kappa(z)}} = \frac{L_n}{L_n - z}, \quad (3.48)$$

$$U_{\kappa(z)} = U_{\kappa(z=L_n)} \frac{L_n - z}{L_n} = \frac{2q_R r_l^{\kappa} L_n}{r_l^{\kappa} \rho_{\kappa} \Pi(r_l^{\kappa} - r_0^{\kappa})} \frac{L_n - z}{L_n}. \quad (3.49)$$

The ~~conservation~~ total integral equation of the conservation of energy, mass and momentum in evaporator ~~evaporizer~~ is equal to

$$\begin{aligned}
 & - \int_{R_z=L_n}^{R_z=0} 2\sigma \frac{dR}{R^2} - K_{1n} \frac{2q_n r_l^k L_n}{\rho_n (r_l^{n^2} - r_0^{n^2}) r' L_n} \int_{L_n}^0 (L_n - z) dz = \\
 & = \frac{4q_n^2 r_l^{k^2} L_n^2}{\rho_n \Pi^2 (r_l^{n^2} - r_0^{n^2})^2 r' L_n^2} \int_{L_n}^0 \frac{d(L_n - z)^2}{dz} dz. \quad (3.50)
 \end{aligned}$$

The pressure differential on liquid at the length of the evaporator ~~is equal to~~ L_n is equal to $\frac{2\sigma}{R_{min}} - \frac{2\sigma}{R_l^0}$, and at the length of the capacitor ~~is equal to~~ L_n is equal to $\frac{2\sigma}{R_l^0}$:

$$\begin{aligned} \frac{2\sigma}{R_{min}} - \frac{2\sigma}{R_l^0} - K_{l,m} \frac{q_n r_l^0 L_n L_n}{\rho_m (r_l^0 - r_0^0)^2 r} = \\ = \frac{2q_n^2 r_l^0 L_n^2}{\rho_m r^2 \Pi^2 (r_l^0 - r_0^0)^2}, \end{aligned} \quad (3.51)$$

$$\frac{2\sigma}{R_l^0} + K_{l,m} \frac{q_n r_l^0 L_n^2}{r \rho_m (r_l^0 - r_0^0)^2} = - \frac{2q_n^2 r_l^0 L_n^2}{\rho_m r^2 \Pi^2 (r_l^0 - r_0^0)^2}. \quad (3.52)$$

Let us add obtained equations (3.51) and (3.52) and let us substitute in them (3.46):

$$\begin{aligned}
\frac{2\sigma}{R_{\min}} + \frac{q_K r_l^K L_K K_2 \ln(r_0^K/r_l^K) \mu_K}{r' \delta \rho_K} - K_1 \mu_K \frac{q_K r_l^K L_K L_K}{\rho_K r' (r_l^{K^2} - r_0^{K^2})} + \\
+ K_1 \mu_K \frac{q_K r_l^K L_K^2}{r' \rho_K (r_l^{K^2} - r_0^{K^2})} = \frac{2q_K^2 r_l^{K^2} L_K^2}{\rho_K \Pi^2 (r_l^{K^2} - r_0^{K^2})^2 r'^2} - \\
- \frac{2q_K^2 r_l^{K^2} L_K^2}{\rho_K r'^2 \Pi^2 (r_l^{K^2} - r_0^{K^2})^2} . \quad (3.53)
\end{aligned}$$

Equation (3.53) it is possible to solve relative to q_K or L_K

L_K :

$$q_K = - \frac{\mu_K r' \Pi^2}{4 r_l^K L_K} \frac{A}{B} + \sqrt{\left(\frac{r' \Pi^2 \mu_K A}{4 r_l^K L_K B} \right)^2 + \frac{\sigma \rho_K \Pi^2 r'^2}{R_{m \ln} L_K^2 r_l^{K^2}} \cdot \frac{1}{B}}, \quad (3.54)$$

where

$$A = \frac{K_2 \ln(r_0^K / r_0^N)}{\delta} - K_1 \left(\frac{L_N}{r_l^{N^2} - r_0^{N^2}} - \frac{L_K}{r_l^{K^2} - r_0^{K^2}} \right),$$

$$B = \frac{1}{(r_l^{N^2} - r_0^{N^2})^2} - \frac{1}{(r_l^{K^2} - r_0^{K^2})^2},$$

$$L_K = \frac{\mu_K \Pi^2 r'}{4 q_K r_l^K} \frac{D}{C} + \sqrt{\left(\frac{\mu_K \Pi^2 r'}{4 q_K r_l^K} \right)^2 \left(\frac{D}{C} \right)^2 - \frac{\sigma \rho_K \Pi^2 r'^2}{R_{m \ln} q_K^2 r_l^{K^2}} \cdot \frac{1}{C}}. \quad (3.55)$$

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Here

$$C = \frac{K_1 r_{1n}^2 \Pi^2}{2q_n r_1^2 (r_1^{2n} - r_1^{2n})} - \frac{1}{(r_1^{2n} - r_0^{2n})^2} - \frac{1}{(r_1^{2n} - r_0^{2n})^2}.$$

$$D = K_2 \frac{\ln(r_0^2/r_1^2)}{\delta} + K_1 \frac{L_n}{r_1^{2n} - r_0^{2n}}.$$

$$L_T = L_n r, \quad (3.56)$$

$$Q_{HT} = -q_n 2\pi r_0^n L_T, \quad (3.57)$$

$$Q_{HT} = -q_n 2\pi r_0^n L_T, \quad (3.58)$$

where n is a number of ~~coaxial~~ ^{tube} elements of coaxial thermal ~~device~~ ^{tube} Q_{HT} -
the total amount of heat, applied into evaporator ~~evaporator~~.

5. Coaxial thermal ~~device~~ ^{tube} with the presence of fluid film on the

surface of the core of ~~capacitor~~ ^{condenser}.

Let us assume that the speed of the motion of liquid in porous core ^{is} insufficiently ~~is~~ great in order to eliminate completely the amount of condensate, which is formed as a result of condensation ^{of} ~~vapor~~ on porous surface. The film of liquid on the surface of core will have ~~that~~ ^{at} thickness ~~with~~ which its thermal resistance will cause increase in the temperature of the surface of film and the retardation of the process of condensation. This will lead to an increase in the total pressure in the vapor phase of thermal ~~film~~ ^{tube} and an increase in saturation temperature both in the zone of condensation and in the zone of evaporation.

Let us examine the process of heat exchange in the zone of condensation when fluid film is present,.

Figure 29b depicts the diagram of porous ~~capacitor~~ ^{condenser} with the presence of fluid film.

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Porous core in the zone of ~~capacitor~~ ^{condenser} has the designations: 1, 2 - fluid film on the surface of the core of ~~capacitor~~ ^{condenser}; 3 - the adiabatic

zone of the core of ~~tube~~ ^{tube}; 4 - porous core in the zone of evaporation;
5 - ~~space~~ ^{vapor} space; 6 - the outer casing of thermal ~~tube~~ ^{tube}; 7 - the inner
shell of thermal ~~tube~~ ^{tube}.

Let us assume that the liquid evenly is exhausted inside porous
core with the speed of $U_{r\kappa}$ on its entire surface
(one-dimensional model of thermal ~~tube~~ ^{tube}). Then for the film of liquid
is valid the notation of the following equations: the equation of
continuity

$$\frac{\partial}{\partial r} (r \rho U_{r\kappa}) = 0, \quad (3.59)$$

the equation of conservation of energy

$$\frac{\lambda_{\kappa}}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) = \rho C_p U_{r\kappa} \frac{\partial T}{\partial r}. \quad (3.60)$$

Boundary conditions:

with $r = r_i^u \quad T = T_{nac}$
 $r = r_0^k \quad T = T_{n,\phi}^k$
 $U_{жк} = U_{r_{жк}}$

Temperature field in fluid film takes the form

$$\frac{T - T_{nac}}{T_{n,\phi}^k - T_{nac}} = \frac{r_0^B - r_i^{nB}}{r_0^{kB} - r_i^{nB}}, \quad (3.61)$$

where $\beta = \rho C_p U_{r_{жк}} \frac{r_{жк}}{\lambda_{жк}}$ - the dimensionless rate of flow of liquid.

Energy balance on (interface liquid - ~~phase~~ ^{vapor}) takes the form

$$r_0^n \lambda_{\text{HK}} \frac{\partial T}{\partial r} \Big|_{r=r_0^K} = r_0^K U_{\text{HK}} \theta_{\text{HK}} r'. \quad (3.62)$$

The thickness of fluid film in that case can be defined as

$$\frac{r_l^n - r_0^K}{r_0^n} = \left[\frac{1}{1 - C_p \frac{T_{\text{max}} - T_{\text{n.б}}}{r'}} \right]^{1/\beta} - 1. \quad (3.63)$$

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Let be assigned ~~prescribed~~ the law of the absorption of liquid in porous core as function of coordinate x (two-dimensional model of thermal ^{tube} ~~cut~~). Let us examine, ^{how} ~~as~~ will depend the thickness of the forming film on coordinate x . Let us assume that thermal energy in the film of liquid is spread by means of thermal conductivity. The

temperature of the surface of fluid film is equal ^{to} T_{nac} . Porous core is made in the form of plate ^{with} the ~~size~~ dimensions of $(L_n + L_a + L_v) bc$.

Then the equation of the preservation of energy will take form

$$\lambda_{\text{m}} \frac{T_{\text{nac}} - T_{\text{n},\phi}}{\delta(x)} = \rho_{\text{m}} r' U_{\text{m}}(x), \quad (3.64)$$

where ~~the~~ $T_{\text{n},\phi}$ - temperature of the surface of porous core.

We now should establish ~~the form of~~ the form of the dependence of the velocity of the absorption of the liquid ~~of~~ U_{m} inside porous core on coordinate x .

Let us examine the process of the motion of liquid in porous body in the form of plate in accordance with the law of Darcy.

$$j_{\text{m}} = - \frac{KA}{\mu_{\text{m}} L} [P_1 - P_2] \quad (3.65)$$

or differentially

$$j_m = - \frac{K}{\mu_m} \nabla P_m. \quad (3.66)$$

In this notation of the law of Darcy, ~~it~~ is disregarded ~~by~~ the effect of gravitational field on the process of the transfer of liquid. If in thermal ^{tube} ~~duct~~ the thickness of porous core is considerably shorter than a radius ~~of the~~ R_n of steam cylindrical channel, then the relationship ^{S₂} ~~obtained~~, obtained for a plate, remain valid.

Work [63] examines the case of the filtration motion of liquid in porous body in the form of plate under the action of pressure gradient. The law of conservation of mass requires in order that

under the stationary conditions of ~~the~~ flow of the incompressible Newtonian liquid would be observed conditions

$$\nabla^2 P_{\text{ж}} = 0. \quad (3.67)$$

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Boundary conditions are the following:

$$y = c, \quad \frac{\partial P_{\text{ж}}}{\partial y} = 0 \quad \text{for all } x,$$

$$y = 0 \quad \left\{ \begin{array}{l} P = P_{\text{ж}}^{\kappa} \\ \frac{\partial P_{\text{ж}}}{\partial y} = 0 \\ P_{\text{ж}} = P_{\text{ж}}^{\text{н}} \end{array} \right. \quad \begin{array}{l} x \geq 0, \\ -L_a \leq x < 0, \\ \text{for } x \leq -L_a. \end{array} \quad (3.68)$$

The diagram of porous core is shown in Fig. 29b.

The solution to the equation of conservation of mass with boundary conditions takes the form

$$U_{\text{ж}}(x) = U_{\text{ж}}(x, 0) = \frac{\pi}{4} \cdot \frac{K}{c} \cdot \frac{\Delta P_{\text{ж}}}{\mu_{\text{ж}}} \cdot \frac{1}{M(\alpha)} \times$$

$$\times \frac{\sqrt{2} \cos(a/2)}{[\cos(\chi + a) - \cos a]^{1/2}}, \quad (3.69)$$

where

$$\chi(\alpha) = \frac{\pi X}{c}, \quad a = \frac{\pi L_a}{2c}, \quad \alpha = \operatorname{tg}(a/2),$$

$M(\alpha)$ - first-order complete elliptic integral with ~~modulus~~ modulus α .

~~General~~ Total fluid flow on porous core is equal to

$$j_m = \int_0^{\infty} \rho_m U_m(x) b dx =$$

$$= \frac{\rho_m b K (\Delta P_m)}{2\mu_m} \cdot \frac{M(\sqrt{1-\alpha^2})}{M(\alpha)} \quad (3.70)$$

Total amount of heat Q , transferred along the thermal ~~tube~~ ^{tube:}

$$Q = j_{\text{ж}} r' = \frac{\pi}{4} \cdot \frac{\rho_{\text{ж}} r' b K (\Delta P_{\text{ж}})}{\mu_{\text{ж}}} \cdot \frac{1}{M(\alpha)} \quad (3.71)$$

under the assumption that $\alpha \ll 1$, $L_a \gg c$. The rate of flow of liquid as function of coordinate x takes the form

$$U_{\text{ж}}(x) = \frac{Q}{\rho_{\text{ж}} r' b c} (\exp x - 1)^{-1/2}. \quad (3.72)$$

The thickness of fluid film can be determined from equations (3.46) and (3.72) in the form

$$\lambda_{\text{ж}}(T_{\text{нае}} - T_{\text{п.ф}}) = \delta(x) \rho_{\text{ж}} r' U_{\text{ж}}(x), \quad (3.73)$$

$$\delta(x) = \frac{\lambda_{\text{ж}}(T_{\text{нае}} - T_{\text{п.ф}}) \sqrt{\exp x - 1} bc}{Q}. \quad (3.74)$$

If one assumes that in evaporator ~~evaporizer~~ the heat removal takes place from the unit of surface evenly and is valid the notation of heat transfer rate in the form

$$\frac{Q}{L_{\text{п}} b} = \lambda_{\text{ф}} \frac{T_{\text{п}} - T_{\text{нае}}}{c}, \quad (3.75)$$

then

$$\delta(x) = \frac{\lambda_{\text{ж}}(T_{\text{нае}} - T_{\text{п.ф}}) \sqrt{\exp x - 1} c^2}{\lambda_{\text{ф}}(T_{\text{п}} - T_{\text{нае}}) L_{\text{п}}}. \quad (3.76)$$

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6. Experimental study of the work of coaxial thermal duct.

In low-temperature laboratory of institute heat- and mass exchange the A.S. of the B.S.S.R was conducted the study of the parameters of the coaxial thermal ducts in which as heat-transfer agent was utilized the ethyl alcohol.

Figure 30 shows experimental ^{diagram of} installation for the study of coaxial thermal tube. The cut ~~section~~ of coaxial tube and the arrangement of sensors are shown in Fig. 29a.

During experimentation on the study of the process of the transfer of thermal energy in coaxial duct it was carried out the recording of the following parameters: 1) the total amount of heat, applied to the external surface of the evaporator/~~vaporizer~~ of the thermal duct Q; 2) the pressure of saturated ^{vapor} ~~liquid~~ within duct; 3) temperature field ^{through} ~~in~~ the thickness of ^{wick} ~~core~~ in evaporator/~~vaporizer~~ and condenser/~~capacitor~~, and also in the vapor phase of duct with the aid of differential copper-constantan thermocouples with the size/~~dimension~~ of thermoelectrodes 0.2 mm and of copper resistance thermometer in the external surface of porous ^{wick} ~~core~~ in the evaporator/~~vaporizer~~ of thermal duct; 4) the flow rate of the cooling fluid, washing the external surface of the condenser/~~capacitor~~ of thermal duct, and temperatures.

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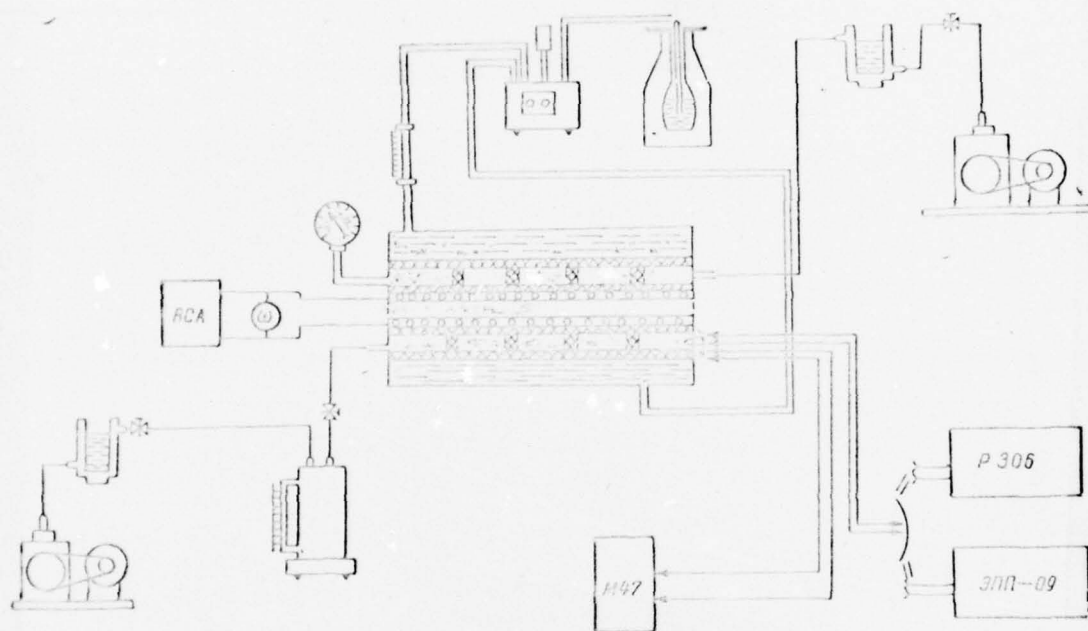
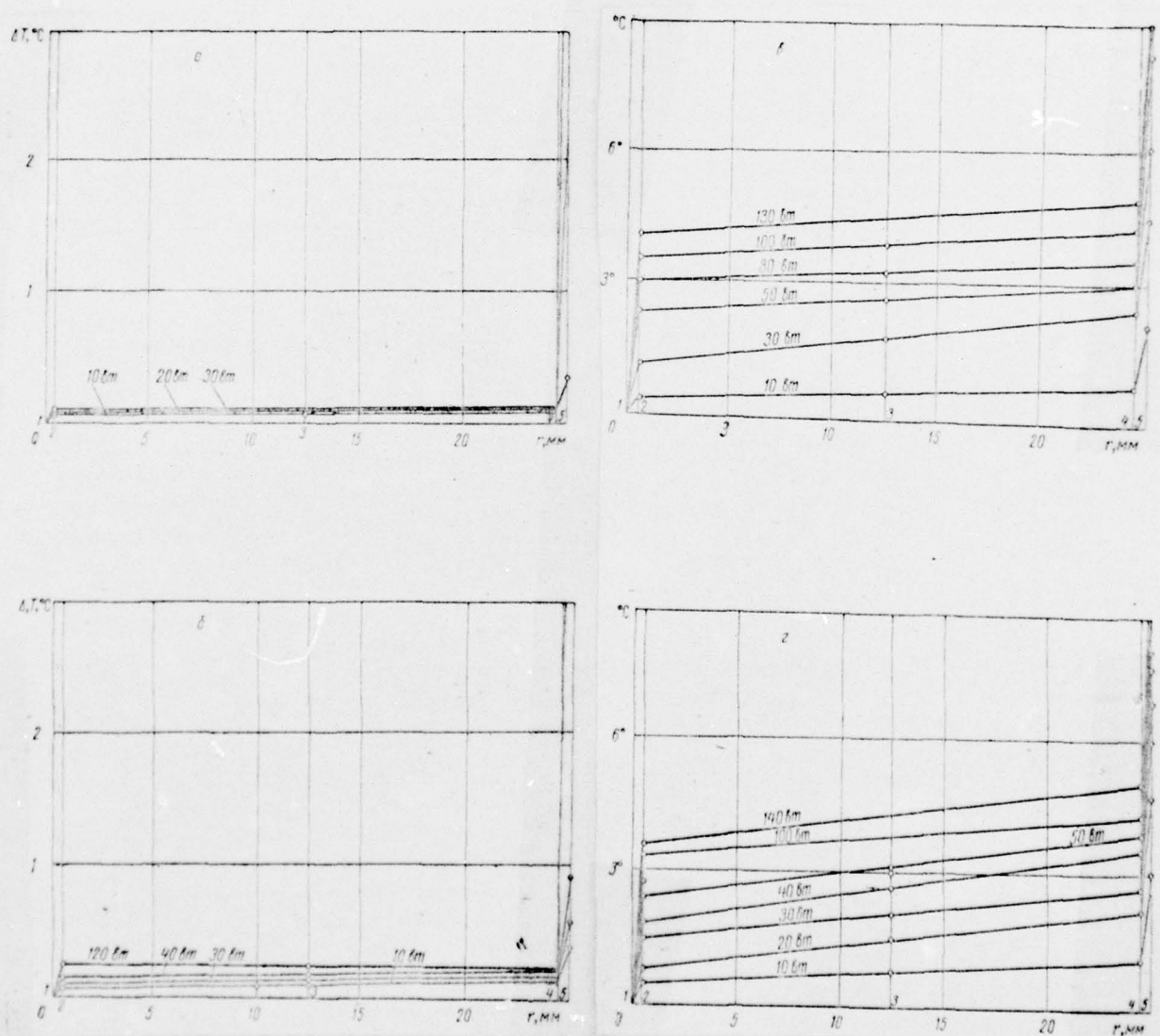


Fig. 30.

Fig. 30. Experimental installation on the study of coaxial thermal duct.

Fig. 31. Temperature distribution in the ~~interval~~/gap between evaporator/~~vaporizer~~ and condenser/~~capacitor~~ of coaxial thermal duct with the volume of the working fluid: a) 10 ml; b) 25 ml; c) 100 ml; d) 200 ml; (1-2) the thickness of condenser (4-5) the thickness of evaporator/~~vaporizer~~.



With the filling of ~~core~~ ^{wick with} 200 ml of liquid (Fig. 31d) the maximum amount of heat, transferred along duct, exceeded 100-120 W. Thus, an increase of the amount of liquid from 100 to 120 ml virtually did not improve, but it impaired the parameters of the work of duct, since ~~the increased~~ the difference of the temperatures ΔT in evaporator/~~vaporizer~~.

As can be seen from figures, temperature ~~drop on pair~~ ^{differential for vapor} it composed 1-2°C; this value in general was overstated, since sensors were located on the surface of ~~core~~ ^{wick}. The basic thermal resistance in coaxial thermal duct falls on ~~core~~ ^{wick} in the region of evaporator/~~vaporizer~~ and in the region of capacitor.

Transformation ratio is calculated from the following formulas through the densities of the thermal flux:

$$K_T = \frac{q_K}{q_H} \approx \frac{A_H}{A_K} \quad (3.77)$$

or, utilizing heat transfer through the ~~core~~ ^{wick} for the saturated ~~core~~ ^{wick}:

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All parameters, necessary for the calculation of the mode/~~conditions~~ of the work of thermal duct, are given below. Figure 31a, b, c, d shows the temperature fields ^{taken with} ~~removed during~~ transmission along the thermal duct of the different amount of thermal energy, during determination in the duct of the different amount of heat-transfer agent.

Figure 31a shows that with the filling of ^{wick with} ~~core~~ (10 ml of alcohol in the evaporator/~~vaporizer~~ of thermal duct was observed the crisis of boiling with the transferred amount of heat $Q = 20-30$ W, i.e., $q_{cr} \approx 790$ W/m².

With the filling of ^{wick with} ~~core~~ (25 ml of alcohol (Fig. 31b) the crisis of boiling was observed during the transmission of heat flux between values $1590 < q_{cr} < 4770$ W/m². Duct managed well with the transmission of the heat flux $q = 1590$ W/m², which corresponded to total rate of heat transmission 40 W.

With the filling of ^{core with} ~~core~~ (100 ml of liquid (Fig. 31c) the crisis of boiling began during the transmission of heat flux between values $4700 < q_{cr} < 5100$ W/m². Duct managed well with the transmission of heat 100 W.

$$K_T^* = \frac{q_k}{q_n} = \frac{\Delta T_k}{\Delta T_n}. \quad (3.78)$$

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Results of experiments with coaxial thermal duct.

The effective length of coaxial thermal duct. 400 mm

Thickness of core in zone c.

the evaporation 0.20 mm

the condensation 0.8 mm

adiabatic.	1.5 mm
Distance between adiabatic zones.	28 mm
Number of cylindrical porous bushings.	4 mm
Length:	
the evaporator/ vaporizer	400 mm
capacitor. condenser	400 mm
Inner diameter:	
the evaporator/ vaporizer	19.5 mm
capacitor. condenser	70 mm
The wall thickness of the housing:	
in the evaporator/ vaporizer	0.25 mm
in capacitor. condenser	1 mm

Thermal conductivity of the material of housing.

20 W/m-°C

Effective thermal conductivity of the ~~core~~^{wick}, filled by alcohol. 1.2 W/m-°C

Thermal conductivity of ethyl alcohol, °C:

0

0.158 kcal/m-h °C

50

0.152 kcal/m-h °C

100

0.147 kcal/m-h °C

Porosity of ~~core~~^{wick}.

70 %

Permeability of ~~core~~^{wick}.

$6 \cdot 10^{-5} \text{ cm}^2$

The maximum ~~altitude~~^{height} of capillary elevation.

4 cm

Minimum radius of meniscus.

0.3 mm

Size/~~dimension~~ of the cell of ~~core~~^{wick}.

0.15 x 0.15 mm

Diameter of the wire of ~~core~~ ^{wick}

0.1 mm

Amount of heat, transmitted along duct, maximum.

140 W

Temperature on the external wall of capacitor.

20 °C

Pressure of saturated pair in duct.

 $f(Q)$

Density is pair, °C: .

17

0.086 kg/m³

50

0.463 kg/m³

Density of liquid, °C: .

17

0.81 kg/m³

50

0.78 kg/m³

Ductility/toughness/viscosity is pair, °C: .

0

78.5 · 10⁻⁶ g/s · cm

100

108 · 10⁻⁶ g/s · cm

Viscosity of liquid, 18°C.

0.122 · 10⁻² kg/s · cm

Heat flux in the zone:

the evaporation

140 W

condensation.

140 W

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Speed:

the evaporation

$$6.27 \cdot 10^{-4} \text{ g/cm}^2 \cdot \text{s}$$

sound with 97°C.

$$269 \text{ m/s}$$

Reynolds number for pair Re

$$f(Q)$$

ΔP_n calculated

$$10^{-6} \text{ atm.}$$

ΔT_n calculated.

$$10-15^\circ \text{C}$$

Heat of vaporization of alcohol.

$$919.6 \cdot 10^7 \text{ erg/g.}$$

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These two coefficients ΔP_n and ΔT_n will agree well between themselves in subcritical mode ~~conditions~~ with the optimum filling

with working fluid - 100 ml.

The preliminarily effective thermal conductivity of porous ~~core~~ ^{wick} can be calculated from formula (3.19).

In conclusion it is necessary to note that when using just one thermal transformer as concentrator and deconcentrator of heat flux the operating temperature pair in the first case is considerably higher.

7. Controlled thermal ducts and the steam chambers.

In a number of cases it is necessary to ~~realize~~/accomplish control of the thermal resistance of ducts not only by ~~the path of~~ the assignment of boundary conditions on their external surface, but also with the aid of other factors, such, for example, as creation of the supplementary diffusion resistance to penetration of the vapors to the surface of condensation with the aid of cushion from non-condensable gas, by means of the artificial turbulization of flow ~~of vapor,~~ ^{pair,} the induced convection of liquid in evaporator ~~vaporizer,~~ effect on the process of the transfer of liquid with the aid of magnetic, ultrasonic, electric fields, vibration or centrifugal

fields, a change in the geometric characteristics of thermal ducts, etc.

Let us examine some of the methods of control of thermal ducts.

Thermal ducts with the use of a cushion of non-condensable gas are utilized as devices for thermostatic control of any objects, since in them is provided the temperature constancy on the larger part of its surface [25].

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Usually thermal ducts with the presence of the uncondensed gas have the supplementary reservoir, in which is accumulated this gas. In the nonoperating state of thermal duct the non-condensable gas on the level with the vapors of liquid is evenly distributed ^{through} ~~by~~ entire volume of ^{vapor} ~~steam~~ space. After the supply of heat flux to the evaporator/~~vaporizer~~ of the thermal duct ~~of~~ the ^{vapors} ~~pair~~ of heat-transfer agent ~~they~~ push aside non-condensable gas into the zone of the condensation, where is formed the interface vapor - gas, which divides capacitor ^{into} ~~by~~ two parts. The temperature constancy of thermal duct is ensured because of the fact that the pressure the ^{vapor} ~~pair~~ of liquid in ^{operating state} ~~running order~~ is subordinated to the exponential law of dependence of P on T on equilibrium curve vapor-liquid, and the

pressure of non-condensable gas is subordinated to linear law for a perfect gas. A change in the temperature in the evaporator/~~vaporizer~~ of thermal duct leads to pressure change ^{of vapor,} ~~pair,~~ the latter, for example, compresses noncondensing gas, is ~~free/released~~ ^{freed} part of the area of capacitor, which leads to an increase in the area of condensation, and therefore to a temperature decrease of duct. If we ~~with any path~~ ^{in some way} change the pressure of non-condensable gas (for example, by means of reduction or increase in the volume of capacitor), then this will lead to a change in the diffusion resistance of thermal duct.

Let us examine the one-dimensional process of condensation ^{of vapor} ~~pair~~ in the capacitor of thermal duct in the presence of non-condensable gas [85]. Let the thermal duct have a length L_T . The temperature in the zone of the evaporation of duct is equal to T_n , and in the zone of condensation on the surface of ^{wick} ~~core~~ T_n . Let us assume that ^{vapor} ~~pair~~ ^{is} transferred through the plane layer of non-condensable gas to the surface of ~~the~~ condensation, for example, of the ^{vapor} ~~steam~~ chamber, with the area of condensation S by diffusion. Then flow ^{of vapor} ~~the pair~~ through the layer of non-condensable gas is equal to

$$j_n = - \frac{DM_1 P}{(P - P_1) RT} \cdot \frac{dP_1}{dy}, \quad (3.79)$$

where P - total pressure in ~~steam~~ ^{vapor} space; P_1 is partial pressure ^{of vapor} ~~pair~~;
 D - coefficient ~~is the~~ ^{of} diffusion coefficient ~~the~~ ^{of} ~~pair~~ ^{vapor} through gas; M_1
 is a molecular weight ^{of vapor} ~~pair~~;

$$D = 0,217 \cdot \left(\frac{P_0}{P} \right) \left(\frac{T}{T_0} \right)^{1,88} \quad (3.80)$$

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Integration of this equation from P_1 with $y = 0$ to P_1 with $y = L_T$ gives

$$\frac{P - P_1}{P - P_{1e}} = \exp \frac{j_n R T L_r}{D_{1n} P} \quad (3.81)$$

If we accept the averaged pressure of non-condensable gas $P_{2cp} = P - P_{1cp}$ and D from equation (3.80), then

$$P_{2cp} = (P - P_{1e}) \frac{j_n T_{cp}^{0.88} 0.217 P_0}{j_n R L_r T_0^{1.88}} \quad (3.82)$$

In the right side of equation (3.82) we disregard the member

$$\left[\exp \left(\frac{j_n R L_r T}{D_{1n} P} \right) - 1 \right]$$

as a result of ~~his~~ ^{its} smallness.

The thickness of the layer of non-condensable gas can be found from the relationship ~~ratio~~, valid for the perfect gas

$$\delta = \frac{P_{2, \text{cp}}}{P} L_T. \quad (3.83)$$

If expression (3.83) is substituted into formula (3.82), then we will obtain flow value ~~the pair~~ ^{of vapor} through the layer of the non-condensable gas

$$i_n = - \frac{(P - P_{1,0}) M_1 C T_{\text{cp}}^{1/2}}{P R \delta_{\text{nr}}}, \quad (3.84)$$

pressure p ~~it~~ corresponds ^{to} T_n , pressure $P_{1,0}$ - to temperature T_K .

If the area of ~~capacitor~~ ^{condenser}, above which is located the cushion of non-condensable gas is equal to S_1 , then the amount ~~pair~~ ^{vapor} that passes per unit time to the surface of condensation ~~equally~~ ^{equals}

$$I_n = i_n S_1. \quad (3.85)$$

The amount of heat, transferred during condensation ^{of vapor}~~air~~, is equal to

$$Q_1 = -j_n [r' + C_p (T_n - T_w)] \approx -j_n r'. \quad (3.86)$$

Coefficient of the heat exchange

$$\alpha_1 = \frac{Q_1}{S_1 (T_n - T_w)} = \frac{P - P_{t_2}}{S_1 P} \times \\ \times \frac{M_1 C T_{cp}^{1/2} [r' + C_p (T_n - T_w)]}{\delta_{n,r} R (T_n - T_w)}. \quad (3.87)$$

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If in the absence of non-condensable gas the thermal resistance of thermal duct is determined by the sum of the thermal resistance of porous ~~core~~^{wick} of the zone of evaporation and condensation R_{rep}

$$R_{\text{rep}} = \lambda_{\text{эф}}^{\text{в}} \frac{T_{\text{н}}' - T_{\text{н}}}{C} + \lambda_{\text{эф}}^{\text{к}} \frac{T_{\text{н}} - T_{\text{н}}}{C}, \quad (3.88)$$

then
~~that~~ in the presence of the cushion of non-condensable gas to this
 sum is added ^{again} ~~as~~ even the thermal resistance of the layer of
 non-condensable gas and the total thermal resistance equals

$$R_1 = R_{u,r} + R_{rep}. \quad (3.89)$$

The amount of heat, transferred along the thermal duct whose
~~condenser~~
~~Capacitor~~ is partially filled by non-condensable gas, it ~~is~~ equals

$$Q = R_1(T_n - T_w)S_1 + R_{rep}(T_n - T_w)S_2, \quad (3.90)$$

$$S = S_1 + S_2.$$

When S_1 ~~it~~ approaches S ,

$$Q = R_1 (T_n - T_R) S, \quad (3.91)$$

when S_1 ~~it~~ vanishes,

$$Q = R_{rep} (T'_n - T_R) S. \quad (3.92)$$

If we equate expression (3.91) and (3.92), then

$$R_1 (T_n - T_R) = R_{rep} (T'_n - T_R), \quad (3.93)$$

$$\frac{T'_n - T_R}{T_n - T_R} = \frac{R_1}{R_{rep}} = \frac{R_{nr} + R_{rep}}{R_{rep}} = \frac{R_{nr}}{R_{rep}} + 1. \quad (3.94)$$

Thus, the presence of non-condensable gas leads to the fact that during the transmission of the identical amount of heat as compared with the case of the absence of non-condensable gas the temperature in the zone of the evaporation of ~~core~~^{wick} must increase, which will cause a pressure increase P within the ~~steam~~^{vapor} chamber. The content of non-condensable gas in the usual thermal ducts and the ~~steam~~^{vapor} chambers is undesirable factor, since increases their thermal resistance.

8. Thermal ducts with worm conveyor - artery.

In this work is proposed one of the methods of a decrease in the thermal resistance of the layer of non-condensable gas by the artificial agitation of flow ~~pair~~^{of vapor} with the aid of its torsion [22, 23].

1. The characteristic feature of low-temperature thermal ducts is the fact that they usually work in the mode/~~conditions~~ of laminar flow ~~pair~~^{of vapor} in the ~~steam~~^{vapor} space of duct. With this axial and radial criterion Re_{pair} for ~~pair~~^{vapor} do not exceed 300. The mode/~~conditions~~ of viscous motion ~~pair~~^{of vapor} in low-temperature thermal ducts occurs on the strength of the fact that the utilized liquids (Freon, water, ammonia, alcohols, cryogenic liquids) possess substantially the smaller ^{latent} heat of vaporization in comparison with metals, lower thermal conductivity, etc, that it does not make it possible to transport along thermal ducts large heat fluxes. Consequently, ~~and~~^{also} the speed of motion ~~pair~~^{of vapor} in them is much lower than in liquid-metal ducts.

It is known that the processes ^{of} heat- and mass exchange in turbulent boundary layers are more intense than in laminar. From this viewpoint ^{is} tempting ^{to} cause artificial agitation ~~pair~~^{of vapor} in low-temperature thermal ducts and thereby to increase the intensity

of heat removal in evaporator/~~vaporizer~~ and heat emissions in
~~condenser.~~
~~capacitor.~~

2. Turbulent mixing ~~pair~~ ^{of vapor} in the ~~range~~ ^{region} of condensation makes it possible to intensify the process of condensation in the presence of non-condensable gas, if it randomly ~~reenter~~/shows within thermal duct, since the non-condensable gas ^{is} pushed aside from the surface of condensation.

3. The torsion of flow ~~pair~~ ^{of vapor} in evaporator/~~vaporizer~~ makes it possible to improve heat removal by boiling, since contributes to separation the ~~pair~~ ^{vapor} also ~~at~~ the drops of liquid, which are formed during boiling and ejected together with vapor from ~~core~~ ^{wick}. The drops of liquid by centrifugal forces again are ~~reject~~/thrown to porous surface, that contributes to an increase in the dryness ^{of vapor} ~~the pair~~ to 99o/o at output/~~yield~~ from evaporator/~~vaporizer~~ and to a uniform wetting of porous surface. The torsion of flow ~~pair~~ ^{of vapor} in evaporator/~~vaporizer~~ stabilizes an increase in the bubbles in porous ~~wick~~ ^{wick} ~~core~~, contributes to their compression and intensifies surface evaporation ~~of~~ ^{from} porous ~~core~~ ^{wick}.

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A deficiency/~~lack~~ in the work of thermal tube with twisted flow

is the fact that somewhat increases the pressure differential in vapor phase ΔP_v , however this insignificant increase ~~it~~ does not play significant role, since $\Delta P_m \gg \Delta P_v$.

In works [22, 23] is suggested the realization of the torsion of flow ~~pair~~ ^{of vapor} with the aid of the worm conveyor, inserted inside thermal duct. If we into the ~~steam~~ ^{vapor} space of thermal duct place hollow metallic worm conveyor with the variable space of torsion, then it is possible to attain an increase several times of the speed of motion of ~~pair~~ ^{vapor} relative to porous ~~core~~ ^{wick} because of the torsion of flow ~~pair~~ ^{of vapor} on the blades of worm conveyor. The twisted nature of flow ~~pair~~ ^{of vapor} creates in evaporator/vaporizer and capacitor the artificial agitation of flow with the aid of centrifugal forces, it increases the intensity of evaporation and condensation.

The worm conveyor, inserted inside tube, plays the role not only baffle ~~the pair~~ ^{of vapor and} also of the stimulator of the process of evaporation and condensation. It can be used as the rigid framework ~~body~~, to ~~fin~~ edges of which is fastened porous ~~core~~ ^{wick}. The production of the housing of worm conveyor in the form ^{of} gently ^{sloping} ^{lights} of cone ~~facilitates~~ its weight and makes it possible to utilize space within worm conveyor for the location there of insert ~~pushing~~ ^{wick} - porous ~~core~~ as supplementary capillary pump (ceramic metal, fiberglass, etc).

The presence of the contact of this supplementary capillary pump with the center section of the evaporator ~~vaporizer~~ of thermal duct is especially important, since precisely this zone of ~~core~~^{wick} is most subjected to the threat of drying with intense heat supply.

Worm conveyor itself can be made from monolithic metal (aluminum, ~~the~~ copper, stainless steel) ~~either~~^{or} from porous ceramic metal or fiberglass.

Figure 32 shows ~~is~~ thermal ~~the~~ duct, which consists of worm conveyor 1, the thin-walled housings of duct 2, of porous ~~core~~^{wick} in the form of wire gauze on the blades of worm conveyor 3, ~~of~~^{wick} porous ~~core~~ within worm conveyor 4.

Thermal duct was made made of ~~the~~ stainless steel, it had length 190 mm and inner diameter 39 mm. The thickness of duct, walls of housing was 0.3 mm.

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As ~~core~~^{wick} on the walls of duct was used the oxidized grid ~~out~~ of the stainless steel ~~with~~ the size/~~dimension~~ of whose cell is 0.16 mm, and the thickness of whose filament is 0.12 mm, twisted into two layers. The porosity of ~~core~~^{wick} composed 70o/o, the maximum altitude of

capillary elevation 5.1 cm., permeability $K = 1.35 \cdot 10^{-9} \text{ m}^2$.

As baffle ~~pair~~^{of vapor} and arteries for the axial transfer of liquid from the zone of condensation into the zone of evaporation was utilized the worm conveyor with the variable space of torsion whose length is 187 mm and whose diameter is 38 mm.

Power supply to the evaporative part of the duct is realized from the Nichrome heater, wound around the external surface of duct. On the other end, ~~lead~~ of the duct is ~~arrange~~/located the ~~capacitor~~^{Condenser} 85 mm long of the type "duct" in duct, according to which is driven off the water with temperature of 12°C.

During experiments were recorded the power input, temperature field along duct, the flow rate of the cooling fluid and the temperature differential at entrance and exit from the section of ~~capacitor~~^{Condenser}. Temperature measurement conducted with the aid of copper-constantan thermocouples.

In Fig. 33a^{is} shown temperature field along duct with worm conveyor and without worm conveyor during the transmission of heat output along ~~axis~~/axis^{of} 100 and 125 W.

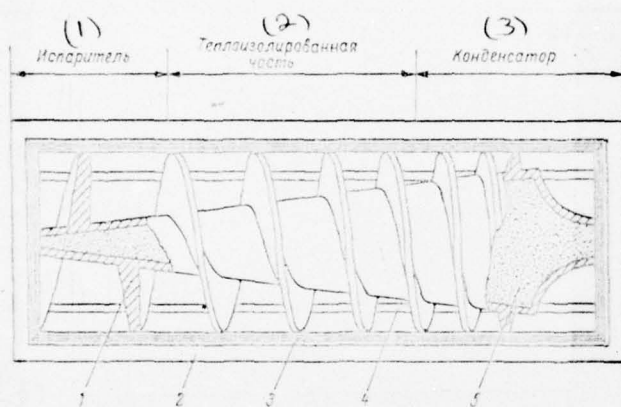


Fig. 32. Thermal duct with the worm conveyor: 1 - worm conveyor; 2 - the housing of duct; 3, 4, 5 - ~~wick~~ ^{wick} ~~core~~.

Key: (1). Evaporator. (2). Heat-insulated part. (3). ^{Condenser} ~~Capacitor~~.

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Figure 33b shows the dependence of the ratio of the thermal resistance of thermal duct without worm conveyor to thermal duct with worm conveyor as function of the transferred along duct heat output.

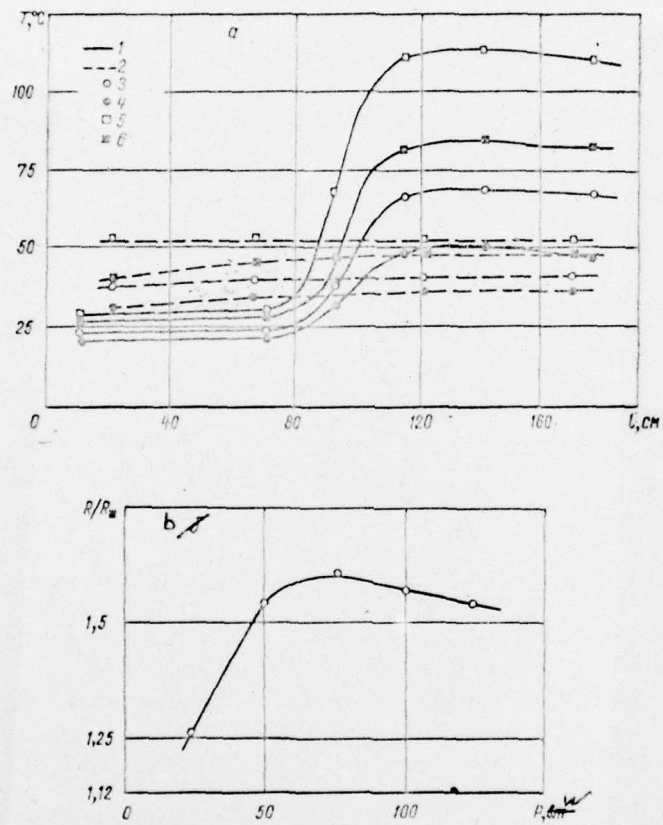


Fig. 33.

Fig. 33. Temperature field along thermal duct with worm conveyor (a); the dependence of the ratio of the thermal resistance of thermal duct without worm conveyor to the thermal resistance of thermal duct with worm conveyor as function of the transferred along duct heat output (b): 1 - temperature field in the core, filled by liquid; 2 - temperature field in vapor phase; 3 - without worm conveyor, $Q = 100$ W; 4 - with worm conveyor, $Q = 100$ W; 5 - without worm conveyor, $Q = 125$ W; 6 - with worm conveyor, $Q = 125$ W.

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The use of a worm conveyor 1.7 times decreased the thermal resistance of thermal duct during the transmission of heat output 75 W.

In conclusion one should say about the fact that the torsion ~~phase~~ ^{of vapor} in cryogenic thermal ducts contributes to an essential decrease in their thermal resistance and to an increase in the transferred heat output.

The twisting of flow ~~pair~~ ^{of vapor} in evaporator, ~~vaporizer~~ and ~~capacitor~~ ^{condenser} is the reliable means for an increase in the coefficient heat- and mass exchange. In works [77, 78] was conducted the comparison of the

intensity of heat exchange during motion in ~~the stand~~^{vertical} pipes of the twisted and axial flows ~~pair~~^{of vapor} with the drops of liquid. The torsion of flow afforded possibility to intensify the process of heat exchange to 200%/o in comparison with axial flow. When axial and twisted flows were compared with an identical pressure differential ΔP , then the local values of the coefficient of the heat exchange of twisted flow were higher by 500%/o.

9. Experimental study of the adjustable thermal ducts.

It is known that during flow ~~the pair~~^{of vapor} through the non-condensable gas can exist three capture modes of the gas: at high speeds and high pressures - the turbulent capture of gas; at low speeds and high pressures - the viscous capture of gas; at low pressures and low speeds - the diffusion capture of gas [112].

During the starting/~~launching~~ of thermal duct occurs the evaporation of liquid into the vapor-gas mixture, equilibrium in which was established ~~installed~~^{at} by diffusion ~~path~~^{method} in the nonoperating state of thermal duct. Beginning from the ~~torque/moment~~ of ~~the~~ time, when rate of evaporation exceeds the speed of concentration and

thermal diffusion, interface gas - ^{vapor} pairs will be gradually shift^{ed} ~~sharred~~ to the side from the zone of the evaporation, where ~~it~~ at the initial moment ^{it} was formed as a result of an increase in the concentration ^{of vapor} ~~pair~~. Analytically this appears as follows:

$$Q/r \geq -\rho \left(D\Delta m_i + \frac{D_T}{T} \Delta T \right). \quad (3.95)$$

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On the value of a change in the position of boundary vapor - gas have an effect of two process.

1. With change of the volumes ^{of the vapor} ~~the pair~~ and gas with the equality of pressures in these volumes, will occur condensation ^{of vapor} ~~pair~~ on the freed from gas surface of ^{wick} ~~core~~. During the process of the condensation of molecules ^{of vapor} ~~the pair~~ is ~~realize~~/accomplished the capture of the molecules of gas whose amount is still sufficiently great in steam space. The character of capture for the majority of the cases can be considered diffusion. The start-up conditions of duct it is possible to consider ^{completed} ~~final~~, when the diffusion capture of gas passes into viscous capture, i.e., when the amount of molecules of gas will become insignificant in ^{vapor} ~~steam~~ zone, in consequence of

which sharply it will increase the coefficient of condensation and, consequently, also rate of ^{vapor} transfer, ~~pair~~.

The reasonings pointed out above are accurate for laminar ^{vapor} flows, ~~pair~~, i.e., $Re < 1000$ (duct of the moderate temperature range). It should be noted that flow conditions ^{of vapor} ~~the pair~~ in thermal duct depends on heat-transfer agent, ~~to~~ transmitted power, the lengths of duct and boundary conditions.

The temperature pair is determined for the most being encountered case - the boundary third-order conditions in cooler and heater according to the following formula:

$$T_n = \frac{Q}{2\pi} \left(\frac{K_n}{S_n} - \frac{K_n}{S_n} \right) - \frac{T_x - T_n}{2}, \quad (3.96)$$

where

$$K_H = \frac{1}{\alpha_1 d_1} + \frac{1}{2\lambda} \ln \frac{d_2}{d_1} + \frac{1}{\alpha_2 d_2},$$

$$K_R = \frac{1}{\alpha_3 d_1} + \frac{1}{2\lambda} \ln \frac{d_2}{d_1} + \frac{1}{\alpha_1 d_2}.$$

2. Effect on the value of a change in the interface vapor - gas exerts ^{also} ~~and~~ the fact that, on one hand, pressure ^{of vapor} ~~the pair~~ in ^{operating} ~~running~~ state follows ^{law} ~~order~~ behaves logarithmic ~~by~~ P from T in the curve of saturation, and, on the other hand, the volume of gas linearly depends on pressure.

For the majority of liquids, dependence $P = f(T)$ is subordinated to the empirical equation of Antoine [112]

$$\lg P = A - \frac{B}{T_n + C} \quad (3.97)$$

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If the equation of Antoine (3.97) we substitute into the equation of state of perfect gas, then we will obtain the dependence of the volume of vapor lock on temperature ~~this~~ *of vapor*

$$V_r = \frac{mRT_r}{M 10^{A - \frac{B}{T_r + C}}} \quad (3.98)$$

For the thermal duct of round cross-section, the length of vapor lock takes the form

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$$L_r = 4 \frac{mRT_r}{M\pi d_n^2} 10^{\frac{B}{T_n+C}-A} \quad (3.99)$$

The area of the zone of condensation, assuming ~~to be~~ $L_a = 0$, is equal to

$$S_n = \pi d_n (L_{rp} - L_{nen}) - 4 \frac{mRT_r}{M d_n} 10^{\frac{B}{T_n+C}-A} \quad (3.100)$$

By solving together equations (3.96) and (3.100), it is possible to obtain the dependences of $T_n = f(Q)$, $L_r = f(Q)$. For the case when in duct it is necessary to ~~support~~ ^{maintain} the constant temperature of heat source, usually are introduced reservoir with gas [113] or pressurization volume [114].

Expression (3.99) for a thermostat with gas meter takes the form

$$L_r' = 4 \frac{mRT_r}{M\pi d_n^2} 10^{\frac{B}{T_n+C}-A} - 4 \frac{V_{p.r.}}{\pi d_n^2}, \quad (3.101)$$

for a thermostat with pressurization volume

$$L_r' = 4 \frac{mRT_r}{M\pi (d_n^2 - d_{n,o}^2)} 10^{\frac{B}{T_n+C}-A} - \frac{d_n^2 L_p}{d_n^2 - d_{n,o}^2}. \quad (3.101')$$

From the analysis of expressions (3.101) and (3.101') it follows that for an increase in the stabilization of temperature it is necessary either to increase or decrease the diameter of duct, and for a thermostat with pressurization volume to decrease the clearance between the pressurization volume and the ^{wick} ~~core~~.

In conclusion of the calculated part, it is necessary to note that analogous calculation can be conducted for any boundary conditions: for gases strongly differing from the ideal it is necessary during the composition of equation (3.98) to use the

equation of van der Waals.

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Description of experiment and experimental ^{device} ~~installation~~.

^{Device} ~~Installation~~ is the thermal duct, manufactured from copper of $\varnothing 12$ x 1, ^{with} ~~by~~ length $L = 300$ mm. ^{Wick} ~~Core~~ consists of three layers of brass grid with parameters of $V = 8.47 \text{ cm}^2$, $\Pi = 0.7$. As working fluid serves 96o/o- ethyl alcohol. As non-condensable gas in the first and third series of experiments, was utilized air $M = 29$, $m = 0.019 \text{ g}$, and in second series of experiments - argon, $M = 39.9$, $m = 0.027 \text{ g}$. Heat flux is created by electrical heater ^{with} ~~by~~ length $L = 7 \text{ cm}$. For the precision measurement of power above the basic heater, is placed guard. As condenser ~~capacitor~~ serves the remaining part of the thermal tube. Heat removal in the first series of experiments was ~~realize~~/accomplished by γ free convection, and in the second and third series - forced; the latter is ~~realize~~/accomplished with the aid of fan $v \approx 10 \text{ m/s}$. Along an entire zone of condensation, they are ~~arrange~~/located 17 copper-constantan thermocouples, caulked into the housing of thermal duct.

The ~~target~~/purpose of the first series of experiments entailed a comparative study of the work of thermal duct without and in the presence of non-condensable gas - air. The results of experiments are

given in Fig. 34.

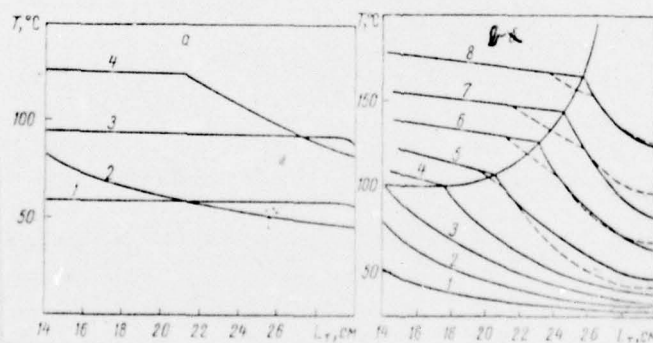


Fig. 34. Distribution of temperature along thermal duct with the non-condensable gas: a) cooling by free convection for powers 8 and

15 W (1 and 3 - without gas, 2 and 4 - with gas); b) cooling by forced convection — experimental, -- calculated; 1 - 11 W; 2 - 28; 3 - 36; 4 - 50; 5 - 70; 6 - 84; 7 - 100; 8 - 130 W.

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Curves 1 and 3 represent the distribution of temperature along the thermal duct, which works in usual mode ~~conditions~~; curves 2 and 4 - the distribution of temperature along the thermal duct, which works in the adjustable mode ~~conditions~~ (i.e. is present non-condensable gas).

The second series of experiments entailed the experimental study of the effect of non-condensable gas - argon on the work of thermal duct. Argon is selected in order to avoid the oxidation processes and electrochemical corrosion, since thermal duct worked at elevated temperatures. Results are given in Fig. 34.

From analysis ^{is} evident (Fig. 34a) that the clear interface vapor - gas appears at any threshold power $8 < Q_{\text{nop}} < 15$ W, which will agree with equation (3.95).

During the analysis of the second series of experiments, were

also checked
~~counted~~ the distribution curves of the temperature in the zone, occupied with non-condensable gas. Reynolds number with blowout ^{*with*} ~~as~~ air whose flow has parameters $v \approx 10$ m/s, $t = 200^\circ$, $Re \approx 8000$, i.e., flow conditions has turbulent character. Nusselt's criterion during turbulent flow conditions takes form $Nu = 0.18 Re^{0.62} \approx 48.5$. For the sake of simplicity in the calculation, we take $d\alpha/dT = 0$, i.e., heat-transfer coefficient does not depend on temperature ($\alpha = Nu \lambda/d \approx 107.2$ W/m².°C). Accepting, that the heat along duct in the zone of vapor lock is spread only by thermal conductivity, we can obtain the distribution of the temperature in the form

$$t = t_0 \frac{\operatorname{ch} m(x-l)}{\operatorname{ch} ml}, \quad (3.102)$$

where

$$m = \sqrt{\frac{\alpha m}{\lambda_f}}.$$

Figure 34b shows that the theoretical and experimental data

somewhat differ from each other ~~themselves~~. Probably this is connected with the fact that is not taken into account the condensation ~~of~~ flow ^{of vapor} ~~the pair~~ through vapor lock, the thermal conductivity of ~~core~~ ^{wick} and heat transfer for gas, or effect ~~of~~ $\alpha = f(T_n)$ and boundary conditions on the end ~~lead~~ of the duct.

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For testing theoretical ~~fining~~/calculations (3.98) and (3.101') were ~~designed~~ ^{calculated} the volume of gas and the length of vapor lock.

Equations (3.98) and (3.99) in the system of SI for ethyl alcohol at the temperature of the vapor lock of $t_p \approx 20^\circ\text{C}$

$$P_r = 0,077 \cdot 10^{\frac{8,42 - \frac{1700}{T_p + 230}}{}} \quad (3.103)$$

$$V_r = 21,37 \cdot 10^{\frac{\frac{1700}{T_p + 230} - 8,42}{}} \quad (3.104)$$

The slope ~~inclination~~ of the section of curves in all experiments is explained by the processes of the gas diffusion in ^{vapor} ~~pairs~~, and, on the contrary, diffusion ^{of vapor} ~~pair~~ into gas explains the overestimate of the course of experimental curves in the zone of vapor lock (Fig. 34b). It should be noted that the slope ~~inclination~~ of curves in operating range will strongly depend on relation

$$K = \frac{M_u}{M_v} \quad (3.105)$$

With $K \gg 1$, will occur the thermal gas diffusion in ~~of pairs~~^{to vapor}, which will lead to an increase in the temperature differential in operating range. Consequently, for a decrease in the $\frac{dT_r}{dL}$ it is necessary to select gas with large molecular weight.

To evaluate the heat stabilization of the adjustable thermal ducts, one should utilize the so-called coefficient of temperature sensitivity

$$\sigma = \frac{dQ}{dT_u} \quad (3.106)$$

The task of the third series of experiments was investigation of the effect of the mass of gas in reservoir on the temperature

sensitivity of the adjustable thermal duct. As non-condensable gas was utilized ~~the~~ air. Mass of gas in reservoir under the normal conditions: $m_1 = 1.13 \cdot 10^{25}$ kg; $m_2 = 3.63 \cdot 10^{-5}$ kg; $m_3 = 6.85 \cdot 10^{-5}$ kg.

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Dependence $T = f(Q)$ is given in Fig. 35b. From the figure one can see that the temperature sensitivity with increase ^{per} $1/m$ increases ($\sigma m_1 = 9$ W/°C, $\sigma m_2 = 2$ W/°C, $\sigma m_3 = 1$ W/°C). The curve of the dependence ^{if} σ on m_p is given in Fig. 35a.

The experimental data showed that threshold power (see Fig. 34a) it lie ~~rests~~ $6 < Q_{\text{top}} < 12$ am.

The results of the given work give sufficient ~~basic~~ bases for using the obtained dependences for the calculation of the adjustable tubes.

10. Thermal ducts, controlled with the aid of centrifugal field and the induced convection of liquid in evaporator/~~vaporizer~~.

In works [34, 35] are described the rotating ^{wickless} ~~coreless~~ thermal ducts, used for cooling the rotors of electric generators, the turbines and other thermally loaded rotating apparatuses. In centrifugal thermal ducts the circulation of heat-transfer agent from the zone of condensation into the zone of evaporation occurs with the aid of centrifugal forces. Figure 36 depicts one of the versions of the use of centrifugal thermal ducts [84, 85].

Centrifugal thermal ducts have a series of advantages in comparison with wick thermal ducts. Basic of them ~~they~~ are: 1) centrifugal ducts at any moment are ready for work; ~~in~~ start-up time ~~it~~ is calculated by fractions of a second; 2) they have less thermal resistance; 3) ^{they} transfer ~~by~~ an order larger heat fluxes per ~~the~~ unit of area; 4) they work well ^{with} ~~during~~ any attitude ^{in space} ~~sensing~~.

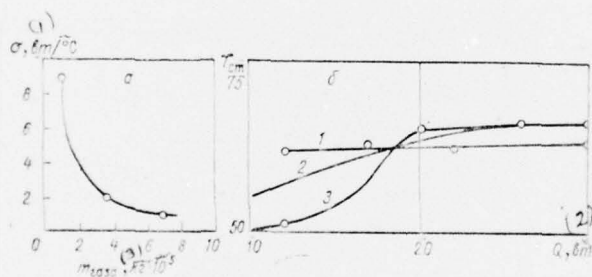


Fig. 35. Dependence a) $\sigma = f(m)$; b) $T_{0T} = f(Q)$: 1 - m_1 ; 2 - m_2 ; 3 - m_3 .

Key: (1). W/°C. (2). W. (3). kg.

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In this work is described one of the constructions of coaxial centrifugal thermal duct [84], intended for the heating of the fluid ^{flow} or gas, that move within coaxial duct. A difference of this thermal duct from the coaxial thermal duct, described in [85], lies in the fact that the pumping of liquid from condenser/capacitor ^{wick} (core tube) to evaporator/vaporizer (external duct) is ~~realize~~/accomplished not with the aid of porous cylindrical inserts, but with the aid of the effect of centrifugation (Coriolis forces) during the rotation of duct around its ~~axle~~/axis.

The proposed construction must ensure the intensification of heat exchange within duct both in the gravitational field and under conditions of weightlessness. As the source of heating it is possible to utilize either flux of radiation or convection current of gas or liquid etc. The ~~target~~/purpose of the application/use of the proposed centrifugal coaxial duct is ^{imparting the energy} ~~energizing~~, conducted to the external surface of duct ^{to} the fluid ~~flow~~ and gas ^{flow}.

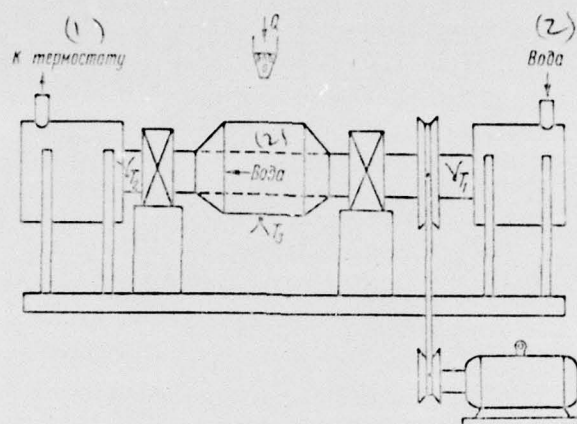


Fig. 36. (Caption next page).

Fig. 36. The diagram of coaxial centrifugal thermal duct in which is utilized the effect of centrifugal field for the return of condensate into the zone of evaporation.

Key: (1). To thermostat. (2). Water.

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Especially is effective the use of similar type thermal ducts for a power supply to them by ~~emission~~^a/radiation in the vacuum when it is not possible to utilize other transmission modes of energy.

This ~~goal~~^{goal}/~~purpose~~ is reached by the fact that as heat exchanger is utilized the coaxial thermal duct, which rotates around its ~~axis~~^a/axis at ~~the~~ definite rate, which makes it possible evenly to wet entire thermally loaded surface within thermal duct with the aid of centrifugal forces and not to utilize for this purpose a porous ~~wick~~^{wick} ~~core~~, i.e., to make a thermal duct without porous ~~core~~^{wick}. The principle of the transmission of heat flux from the hot medium to cold gas or liquid flow entails the following. If thermal duct is made coaxial (duct in duct), is forced ~~it~~ to rotate around its ~~axis~~/axis and along ~~wick~~^{wick} ~~core~~ tube to pass cold flow, ~~but~~^{and} external surface to heat, then ~~is~~

the liquid, which partially fills the space between the external and internal surfaces of thermal duct in which is absent the non-condensable gas, with the aid of centrifugal forces will be pressed against the external wall of thermal duct, after cooling by its course of evaporation. ^{The formed} ~~Generatrix~~ in this case ^{vapor} ~~pairs~~ moves to the internal cold wall of duct, it ^{releases latent} ~~isolates~~ heat of vaporization ^{being-} ~~condensing~~ ^{condensed} on it in the form of the drops which by centrifugal forces again ^{are} ~~reject/throw~~ to the external wall of duct, which makes it possible to heat the flow of gas or liquid, which moves along the internal duct of the rotating coaxial thermal duct. The coefficient of heat exchange between the pipe flow has high value, 5-10 times exceeding the coefficient of heat exchange between the flow and the motionless duct, as a result of the fact that occurs both the directed axial flow and the flow of Couette, formed ^{by} ~~the~~ rotation of the walls of thermal duct and ~~that~~ facilitating the agitation of flow.

Figure 36 shows ^a ~~the~~ diagram of coaxial thermal duct with the use of centrifugal acceleration. The principle of its work entails the following. ^{Through} ~~over~~ duct moves the flow of cold gas or liquid, which must be heated.

The heat flux Q will be fed to external ^{heat-exchanger} ~~radiator~~ surface, and, passing through the wall, produces the evaporation of fluid film,

held on internal surface by centrifugal forces ^{with} ~~during~~ the rotation of heat exchanger.

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^{Vapor}
~~Passes~~ it moves over the intertube space from which is eliminated the non-condensable gas, and it is condensed on cold surface, giving up ^(latent) heat of vaporization to flow r' through the wall.

The forming during condensation ^{of vapor} ~~part~~ liquid in centrifugal-force field again returns to wall 2, and the process of heat transfer ~~it~~ becomes stationary. The concave surface ~~of~~ ^{of} 2 heat exchanger ~~permits~~ implementation of displacement of liquid into the thermally loaded zone, since the component of centrifugal force is directed to the center of concave surface. The evaporation of liquid from thin film on metallic surface in centrifugal-force field makes it possible to remove ~~take~~ the heat fluxes, transferred by an order higher than those which are transferred ones along thermal ducts with porous ^{wick} ~~core~~, since the centrifugal forces which can reach to 1000 g, impede the emergence of bubbles and substantially they shift ~~shear~~ the crisis of boiling to the side of large heat fluxes.

One should indicate the fact that this heat exchanger possesses the diode properties of the transmission of heat flux. It transfers

heat only from external surface to internal and does not transfer from the internal to external because of the action of centrifugal forces. In the presence of hot gas or liquid flow in ~~core~~^{wick} tube and of cold flow on external surface heat will be transferred only by ~~path~~ of thermal conductivity ~~the pair~~^{of vapor} in intertube space, which will ~~compose~~^{comprise} negligibly low value in comparison with heat flux of environment to internal ~~radiator~~^{heat-exchanger} surface, transferred by phase transition (evaporation - condensation).

The experiments conducted with the centrifugal thermal duct, depicted on Fig. 36, are shown in Fig. 37.

Figure 37a shows the dependence of the temperature of the surface of the evaporator/~~vaporizer~~ of centrifugal thermal duct ~~from~~^{on} the velocity of its rotation ~~at~~^{with} constant heat flux. Figure 37b gives the curve/~~graph~~ of the dependence of the temperature differential of liquid coolant at entrance and exit of the condenser/~~capacitor~~ of duct ~~from~~^{on} the temperature of the surface of evaporator/~~vaporizer~~ at the different numbers of revolutions of duct.

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Figure 37c shows the dependence of the temperature of the surface of

the evaporator/~~vaporizer~~ of duct ^{on} ~~from~~ the value ^{of} applied heat flux at the different rotational speeds of duct. As can be seen from this figure, at the rotational speed of duct 47 r/s, it successfully ^{coped} ~~managed~~ with the transmission of heat output ^{of} more than 1 kW; in this case the temperature of the surface of evaporator/~~vaporizer~~ was not more than 82°C, which indicates by no means exhaustable possibilities ^{for} ~~on~~ heat removal in this duct. The temperature of the surface of the evaporator/~~vaporizer~~ of duct was measured with the aid of the special thermocouple, automatically forced ~~against~~ against duct at the ~~torque~~ moment of the cessation of its rotation.

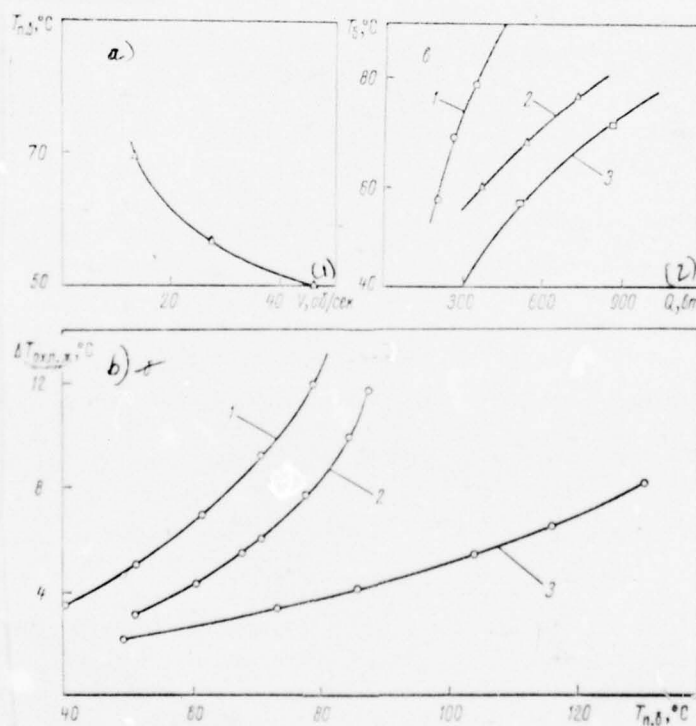


Fig. 37. (Caption next page).

Fig. 37. Dependence of the temperature of the wall of centrifugal thermal duct ~~from~~^{on} the rotational speed (a') and of the temperature of the surface of evaporator/~~vaporizer~~^{on} from the value^{of} applied heat flow (b): 1 - 47 r/s; 2 - 23.5; 3 - 11.5 r/s; (c): 1 - 11.5 r/s; 2 - 23.5 r/s; 3 - 47 r/s.

Key: (1). r/s. (2). W.

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Judging by the obtained experimental results, centrifugal coaxial ducts will have extensive application in a ~~series~~^{number} of ~~branches~~ of industry.

In the centrifugal thermal ducts it is possible to ~~separate~~^{distinguish} three ~~cell~~/elements, on which depends the successful work of the duct: rotating evaporator/~~vaporizer~~ and condenser/~~capacitor~~ and the zone of the transport^{of} two-phase vapor-liquid flow under the action of centrifugal forces. In evaporator/~~vaporizer~~ the heat removal can be ~~realize/accomplished~~^{either in the evaporation mode} or in the mode/~~conditions~~ of boiling.

The process of evaporative cooling can be described with the aid

of GertsA - Knudsen's formula

$$j_n = AS \sqrt{\frac{m}{2\pi KT}} [P - P^*(T)], \quad (3.107)$$

where m - molecular mass; P^* - the pressure of saturated ^{vapors} ~~steams~~ at the temperature of surface; A is a coefficient of evaporation; j_n - the flow of the vaporizing substance; S - area.

The speed of motion ^{of vapor} ~~pair is~~ equals ~~to~~

$$v_r/r=R_n = \frac{AS}{\rho} \sqrt{\frac{m}{2\pi KT}} [P - P^*(T)] = \alpha(T) [P - P^*], \quad (3.108)$$

where the R_n - the inside radius of the surface of evaporator ~~vaporizer~~; ρ is density ^{of vapor} ~~pair~~.

Respectively heat flux from ^a ~~the~~ unit of ~~the~~ surface of evaporator ~~vaporizer~~ is equal to

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$$q = \frac{jr'}{S} = r' \rho v_w \quad (3.109)$$

In the work of evaporator/~~vaporizer~~ in the mode/~~conditions~~ of boiling, the centrifugal forces positively ~~show up in the~~ ^{affect the} process of heat exchange. In works [79, 80] it is indicated the fact that in the rotating boilers is possible to ~~realize~~ ^{with} accomplish heat removal of order $2.5 \cdot 10^3 \text{ kW/m}^2$ ~~during~~ ^{of} acceleration $\Lambda 400 \text{ g}$. In this case, emerging ^{vapor} ~~of pairs~~ has 990/o of dryness. The produced by the action of centrifugal forces angular accelerations lead to the emergence of ~~the~~ induced convection in the liquid film which suppresses the formation/~~education~~ of bubbles.

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This makes it possible to increase the maximum peak loads. Critical heat fluxes with 400 ^{-times} ~~multiple~~ overloads ^{are} $\Lambda 4.5$ times higher than ^{with} ~~during~~ acceleration Λ and can 10 times exceed critical heat flux in thermal duct with capillary-porous ^{wick} ~~core~~.

Experiments [81] showed that the gravitational field, and also centrifugal-force field substantially **affect** the value of the maximum heat flux during boiling the liquid of q_{\max} .

In centrifugal thermal duct the lateral fluid flows substantially affect the value of ~~aaaa~~ ^{g_{\max}} . This is evident from the analysis of the formula of Borishanskiy, ~~the~~ obtained for boiling conditions ^{η} liquid in the large volume:

$$q_{\max} = (0,131 + 4N^{-0,4}) r' \rho_n^{1/2} \sqrt{\sigma g (\rho_m - \rho_n)} \quad (3.110)$$

where

$$N = \frac{\rho_m \sigma}{\mu_m^2} \sqrt{\frac{\sigma}{g (\rho_m - \rho_n)}}.$$

The parameter N characterizes ^{lift} ~~buoyancy effect~~. The value of q_{\max} is determined by the following independent variables of

$\rho_m, \rho_n, r', \sigma, L, g, \mu$. Then it is possible to combine into four groups:

$$\sqrt{1 - \rho_n/\rho_m} \cdot \frac{q_{\max}}{r' \rho_n^2 (\rho_m \sigma g)^{1/4}},$$

$$L \sqrt{g \rho_m / \sigma}, \quad \sqrt{g L \sigma / \mu^2}.$$

Supplementary conversions give the following parameters for determining the effect of the induced convection on q_{\max} :

$$L' = L \sqrt{g \rho_m / \sigma} (1 - \rho_n / \rho_m)^{1/2}, \quad (3.111)$$

$$N = \left(\frac{\sigma}{\mu_m^2} \right) \sqrt{\sigma \rho_m / g} \frac{1}{\sqrt{1 - \rho_n / \rho_m}},$$

$$I = \sqrt{N L'}.$$

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The ^{combination}~~association~~ of the indicated four groups of the parameters makes it possible to find the relation of the maximum heat flux of ^{with}~~during~~ boiling liquid in ^a~~the~~ volume of the finite dimensions with the presence of the induced convection to the maximum heat flux ^{with}~~during~~ boiling liquid on infinite horizontal flat/~~plane~~ plate [82]

$$\frac{q'_{\max}}{q_{\max}} = f(L', I, \sqrt{1 + \rho_n/\rho_{\text{pr}}}). \quad (3.112)$$

According to the formula of Kutataladze value

$$q'_{\max} = 0,131 r' \rho_n^{1/2} \sqrt{\sigma(\rho_{\text{pr}} - \rho_n)} \sqrt{1 + \frac{\rho_n}{\rho_{\text{pr}}}}. \quad (3.113)$$

Thus, in formula (3.112) is considered the effect of scale parameter I and of the parameter of the lift of the induced

convection N on the value of q_{\max} .

As can be seen from formula (3.113), the critical heat flux of q_{\max} depends on acceleration g to degree of $1/4$, according to formula (3.112) it also depends on the geometric dimensions of system. According to the calculations, when the field of accelerations ^{of 1000g} is present, ~~1000g~~ it is possible to achieve the flows of $q_{\max} \approx 5.6 \cdot 10^3$ kW/m². Such accelerations can be obtained during the rotation of duct 6 cm. in diameter with speed ~6000 r/min.

Heat exchange during condensation ^{of vapor} ~~pair~~ in the condenser/~~capacitor~~ of centrifugal thermal duct is more intensive than during condensation on motionless surface, since fluid film has the minimum thickness because of the presence of centrifugal-force field.

The coefficient of heat transfer to wall during condensation of ^{vapor} ~~pair~~ in the condenser/~~capacitor~~ of centrifugal thermal duct can by an order exceed the coefficient of heat transfer by the condenser/~~capacitor~~ of thermal duct with porous ^{wick} ~~core~~. Centrifugal forces break away fluid film in the condenser/~~capacitor~~ of thermal duct and disperse drops ^{over} ~~by~~ the area of evaporator/~~vaporizer~~. The remaining film with the aid of Coriolis forces is moved into the zone of evaporator/~~vaporizer~~.

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CONCLUSION.

It is difficult at present to predict all the possible directions of the further development of works in the construction of thermal ducts. Is too short the history of their emergence. However, with complete confidence it is possible to say that they will find wide use in our to mode of life and in a ^{number} ~~series~~ of the branches of industry.

The guarantee of the successful functioning of thermal ducts must be good theory of their work which still needs at present essential modification, and also ^{worked-out} ~~waste~~ technology of the production of porous ^{wicks} ~~cores~~ and housing of thermal ducts. Industry begins to master the issue of the broad spectrum of metallic porous materials, and also the porous dielectrics, which possess high thermal conductivity. With their appearance, probably, will be feasible new jump in the construction of the thermal ducts and ^{vapor} ~~steam~~ chambers.

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